OIL HYDRAULIC POWER AND ITS INDUSTRIAL APPLICATIONS

Oil Hydraulic Power and Its Industrial Applications

BY

WALTER ERNST, M.E.,
Member, ASME
Hydraulic Consulting Engineer
Vice-President and Director of Engineering,
The Commonwealth Engineering Co. of Ohio
Dayton, Ohio
Formerly Director of Engineering,
The Hydraulic Press Mfg. Co.
Mount Gilead, Ohio

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PREFACE

The basis of the art of oil hydraulics is the science of fluid mechanics. This book is intended to present a history of this art, a textbook and design guide, and a picture of its present state of development. It is hoped that it will be found useful not only for students, and engineers now engaged in design and manufacture of oil hydraulic equipment, but especially for present and potential users of this means of power transmission. To this end, much material has been incorporated dealing with available components and their use and application.

Like all mechanical arts and sciences, oil hydraulics had its forerunners and pioneers, such as H. S. Hele-Shaw, Conrad M. Conradson, Reynold Janney, W. E. Magie, Walter Ferris, to mention only a few. But astonishing as it may sound to many, oil hydraulics as an industry dates back only about twenty-five years. The author has had the unique and outstanding opportunity of having this period coincide approximately with his own activity in this field. Few engineers have been given the chance to assist in the birth of a new industry, contribute to its growth, and help in its development, finally to see it mature into a full-fledged branch of arts and sciences.

The desirability, or rather the necessity, of having a text of this kind available became increasingly apparent during the author's connection with the Commonwealth Engineering Company of Ohio immediately after the close of the Second World War. The Commonwealth Engineering Company was then in the process of building up a rapidly increasing staff of hydraulic design, development and research engineers.

It is a pleasure for the author to acknowledge his indebtedness to the many persons and firms who have assisted, directly and indirectly, in the preparation of this book with their encouragement, cooperation, and the permission to utilize published material. Particularly valuable assistance has been received from R. B. Dow, Bureau of Ordnance, Washington, and the American Institute of Physics; Lewis F. Moody, Princeton, N. J.; R. J. S. Piggott, Pittsburgh, Pa., and the Gulf Research and Development Company; G. F. Nordenholt and *Product Engineering;* Geo. A. Stetson and the American Society of Mechanical Engineers; John W. Greve and *Machine Design;* S. H. Monroe and the Socony-Vacuum Oil Company, Inc.; M. E. Engebretson of The Oilgear Company; L. R. Twyman and J. B. Brown of Vickers Inc.; C. M. Reesey and Hans Ernst of

The Cincinnati Milling Machine Company; Gus F. Koehler of Hydro-Power Inc.; D. V. Smyth of Waterbury Tool Division of Vickers Inc.; F. A. Lewis of Linear Inc.; H. A. Toulmin, President and Chairman of the Board of The Commonwealth Engineering Company; J. M. Hush, Staff Physicist of The Commonwealth Engineering Company, for his valuable assistance in some of the more complex mathematical analyses. All the drawings were made from the author's sketches by Leo T. Kruskamp of Dayton. The author wishes to honor the memory of an old friend, D. R. Francis, formerly Chief Engineer of the Waterbury Tool Company, who contributed material for Chap. VII on Axial Plunger Pumps.

The first six chapters of the book are devoted to a recapitulation of the fundamentals of fluid mechanics, to which much material has been added pertaining in particular to its application to oil hydraulics. In the preparation of these chapters, in particular, Chaps. II, III, IV, and V, the text by Dodge and Thomson, "Fluid Mechanics," has been drawn upon, both as reference for arrangement of much of the material and also for some of the analysis from which many of the fundamental concepts of fluid mechanics were derived. The author is indebted to both authors for their kind permission to make use of this material. To those of the readers who wish to make a more profound study of this fascinating subject, this text is cordially recommended.

The author is indebted to Henry Holt and Company, Inc., and Prof. Geo. E. Russell for their permission to use material from Professor Russell's book, "Hydraulics," in compiling the data given in the first chapter on Fundamental Units of Measurement.

The entire manuscript was typed and prepared for printing by the author's wife, without whose assistance the work would not have been possible.

WALTER ERNST

DAYTON, OHIO
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CONTENTS

LIST OF SYMBOLS

Mass = M slugs

Acceleration of gravity = g feet per sec^2

Force = F lb

Weight = W lb

Specific weight = w lb per cu ft

Width = w in.

Breadth = b in.

Density = ρ slugs per cu ft

Specific gravity = s

Length = L ft or in.

Diameter = D, d ft or in.

Radius = R, r ft or in.

Area = A sq ft or sq in.

Volume = V cu in. or cu ft

Specific volume = v cu ft per lb

Velocity = V, u ft per sec

Angular velocity = ω radians per sec

Speed = n rpm

Speed = S ft per sec

Energy = E ft-lb

Pressure = p psf or psi

Head = h ft

Time = t sec

Temperature = t °F

Shear stress = τ psi or psf

Absolute viscosity = μ lb-sec per sq ft or centipoises

Kinematic viscosity = ν sq ft per sec or centistokes

Reynolds Number = R

Specific heat = q Btu per lb per °F

Capacity = Q cu ft or cu in. per sec, cu in. or gal per min

Efficiency = e

Eccentricity = e in.

Torque = T ft-lb

CHAPTER I

FUNDAMENTAL UNITS OF MEASUREMENT

1. Mass and Weight. For a full understanding of the theory and practice of the hydraulic art, a thorough acquaintance with the fundamental units of measurement as used in the analysis of hydraulic problems is absolutely essential.

Four systems of measurement are currently in use, two English and two metric, varying from each other basically in the selection of the primary units of mass and force. Data in handbooks and tables may be given in units of any of these systems, and a knowledge of their basic units and conversion factors is indispensable.

Any system of measurement must contain three fundamental units of time, length, and mass or force. All other units in the same system may be derived from the fundamental units by combination with one or more of the same or other fundamental units. For instance, cubic measurement is merely a function of the original unit of length, L, being measured in L^3.

Much confusion has been caused by the fact that the names of some of the units of mass and weight have been used interchangeably, and to avoid these difficulties, a careful definition of the terms of mass and weight is given.

Mass. Mass may be defined as the amount of matter in a given body. The amount of mass is an unvarying quantity, independent of location on the earth or any other planet. Masses may be compared with each other by weighing them on a lever balance.

Weight. Weight may be defined as the amount of pull exerted by the earth on a given body. Since the gravitational pull varies with the location on earth, and more so on any other planet, weight is not a constant quantity for any given body.

2. Systems of Measurement. Newton's second law of motion states that a force acting on a body will produce an acceleration, so that the product of its mass times its acceleration will equal the force.

$$F = aM \tag{1}$$

According to whether we consider force or mass as basic unit and derive the other from it by a suitably chosen factor of acceleration, we arrive at the two basic systems in either the English or metric measurement.

In the English absolute system, the mass is used as basic unit. One pound mass by definition is the mass of a cube of platinum carefully preserved in London.

The force that will produce an acceleration of one foot per second per second upon this pound mass is called a "poundal." Therefore the basic units used in the English absolute system are

Unit of mass	1 lb
Unit of force	1 poundal
Unit of length	1 ft
Unit of time	1 sec
Unit of acceleration	1 ft per sec^2

The English absolute system is used by physicists in English-speaking countries.

In the English gravitational system the force is used as a basis. One pound force is defined as the force that imparts an acceleration of 32.174 ft per sec^2 to the standard of mass in the English system, or, in other words, it is the weight of the unit of mass at sea level at 45° latitude. (32.174 is acceleration of gravity at sea level and 45°.)

Since the pound force imparts 32.174 feet per sec^2 to the pound mass, it must impart 1 ft per sec^2 acceleration to 32.174 lb mass. This amount of mass is called "one slug." Therefore, in the English gravitational system, we have

Unit of mass	1 slug
Unit of force	1 lb
Unit of length	1 ft
Unit of time	1 sec
Unit of acceleration	1 ft per sec^2

3. Conversion of Units in Both Systems. We have seen that 32.174 lb mass equals 1 slug. By definition 1 poundal produces an acceleration of 1 ft per sec^2 upon 1 lb mass. Also 1 lb force produces an acceleration of 32.174 ft per sec^2 upon 1 lb mass. Therefore

$$1 \text{ lb force} = 32.174 \text{ poundals}$$

We have seen that 1 lb force is the weight of 1 lb mass at sea level and 45°; therefore 1 lb weight *at sea level* is *numerically* (not dimensionally) equal to 1 lb mass.

If W_0 = weight at sea level,
then

$$M = \text{mass in slugs} = \frac{W_0}{32.174} \tag{2}$$

If W = weight at any other locality,

$$W = \frac{W_0 g}{32.174} \tag{3}$$

$$M = \frac{W}{g}$$

where g = gravitational acceleration at that location. (4)

4. Density, Specific Weight, Specific Gravity. The density is the mass of a substance per unit volume.

$$\rho = \frac{\text{slugs}}{\text{cu ft}}$$

Specific weight is the weight per unit volume.

$$w = \frac{\text{lb}}{\text{cu ft}}$$

$$\rho = \frac{w}{g} \tag{5}$$

Specific gravity is the ratio of the specific weight of a substance to that of an equal amount of water. Water weighs 62.4 lb/cu ft.

The English gravitational system is used almost exclusively in engineering practice in English-speaking countries. Unless otherwise noted, calculations in this book will be based on the English gravitational system.

5. Length, Area, and Volume. Since all measurements of length in both English systems are given in feet, all measurements of area and volume must be given in square feet and cubic feet. For the sake of convenience, distances, areas, and volumes are often given in inches, square inches, and cubic inches and must be converted where necessary.

$$\begin{aligned} 1 \text{ ft} &= 12 \text{ in.} \\ 1 \text{ sq ft} &= 144 \text{ sq in.} \\ 1 \text{ cu ft} &= 1{,}728 \text{ cu in.} \\ 1 \text{ U.S. gal} &= 231 \text{ cu in.} \end{aligned}$$

6. Energy and Power. In the gravitational system energy or work is expressed in foot-pounds; power or duty is expressed in foot-pounds per second.

$$550 \text{ ft-lb per sec} = 33{,}000 \text{ ft-lb per min} = 1 \text{ hp}$$

7. The Mechanical Heat Equivalent. One British thermal unit is the heat required to raise the temperature of one pound of water one degree Fahrenheit.

$$1 \text{ Btu} = 778.57 \text{ ft-lb}$$
$$1 \text{ hp} = 42.385 \text{ Btu per min}$$
$$1 \text{ atmosphere (English)} = 14.69 \text{ psi}$$

8. Hydraulic Horsepower. To pump Q gal of water against a pressure of p psi, work must be done in the amount of

$$\frac{Q \times 231 \times p}{12} \quad \text{ft-lb} \tag{6}$$

If this work must be done in 1 min, the power required will be

$$\frac{Q \times 231 \times p}{12 \times 33{,}000} \quad \text{hp} \tag{7}$$

or

$$0.0005833Qp \quad \text{hp} \tag{8}$$

9. The Metric Absolute System (Cgs System). The mass is used as basic unit. One gram mass is the $\frac{1}{1000}$ part of a platinum cube preserved in Sèvres, France, near Paris. One gram mass also equals the mass of 1 cc of distilled water at 4°C.

The unit of length is the centimeter or $\frac{1}{100}$ part of the length of a platinum bar measured at 0°C, which is kept near Paris. This unit was originally intended to represent one ten-millionth of the earth meridian quadrant, but was subsequently found slightly smaller.

The unit of time is one second or the 1/864,000 part of a mean solar day.

A force that imparts an acceleration of one centimeter per second per second to the mass of one gram is called a "dyne."

$$1{,}000{,}000 \text{ dynes} = 1 \text{ megadyne}$$

In the cgs system, energy is expressed in ergs.

$$1 \text{ dyne-cm} = 1 \text{ erg}$$
$$10^7 \text{ ergs} = 1 \text{ joule}$$

Power or duty is expressed in watts.

$$10^7 \text{ ergs per sec} = 1 \text{ watt}$$
$$1 \text{ kw} = 1{,}000 \text{ watts}$$

The cgs system is used in physics and electrical engineering the world over.

10. The Metric Gravitational System. The unit of force is the kilogram. One kilogram is defined as the force that produces an acceleration of 9.807 m per sec^2 on the mass of one kilogram, or, in other words, it is the weight of one kilogram mass at sea level and 45°C.

The unit of length is the meter.

The unit of mass in the metric gravitational system has not been defined by name.

Energy is expressed in meter-kilograms. Power or duty is expressed in horsepower (metric).

$$1 \text{ hp} = 75 \text{ m-kg per sec}$$

The Mechanical Heat Equivalent. One calorie is the heat required to raise the temperature of one kilogram of water one degree centigrade.

$$\begin{aligned} 1 \text{ cal} &= 427 \text{ m-kg} \\ 1 \text{ hp} &= 10.534 \text{ cal per min} \\ 1 \text{ atmosphere (metric)} &= 1 \text{ kg per sq cm} \end{aligned}$$

The metric gravitational system is used in engineering calculation in those countries which have adopted the metric system of weights and measures.

11. Conversion of Units in the Metric Systems. One kilogram force imparts an acceleration of 9.80665 m per sec^2 to 1-kg mass. One dyne imparts an acceleration of 1 cm per sec^2 to 1 g mass. One kilogram force, therefore, equals 980,665 dynes or almost 1 megadyne.

Since 1 kg weight represents the weight of 1 kg mass at sea level and 45°, 1 kg weight must equal *numerically* 1 kg mass.

TABLE I. CONVERSION BETWEEN ENGLISH AND METRIC SYSTEMS

Quantity	English absolute	English gravitational	Cgs	Metric gravitational
Mass	32.17 lb	1 slug	14,594 g	
Force	32.17 poundals	1 lb	444,822 dynes	0.4536 kg
Length	1 ft	1 ft	30.48 cm	0.3048 m
Energy	....	1 ft-lb	13,558,200 ergs	0.1383 m-kg
Duty	....	1 hp	745,7 watt	1.0139 hp
Heat	....	1 Btu	252 cal	0.252 cal
Pressure	....	1 atmosphere	....	1.033 atmosphere

12. Density, Specific Weight, and Specific Gravity. Density in the metric system is measured in

$$\frac{g \text{ (mass)}}{\text{cc}} = \rho$$

Specific weight is measured in

$$\frac{g \text{ (weight)}}{\text{cc}} = w$$

Specific gravity is the ratio of specific weight of a substance to that of water.

$$\frac{w}{1} = s \tag{9}$$

In the metric system the three units of density, specific weight, and specific gravity are numerically equal.

CHAPTER II

PROPERTIES OF HYDRAULIC FLUIDS

1. Compressibility. All fluids are compressible, even though liquids for many ordinary problems have been considered incompressible. For high-pressure hydraulic work, the compressibility of the operating oil is of greatest importance and must be taken into consideration in the design of hydraulic machinery.

The compressibility of oil may be defined as the specific change in density per unit of pressure. The compressibility decreases with increasing pressure and must therefore be expressed as a differential quotient.

$$C = \frac{d\rho}{\rho\, dp} \tag{1}$$

Dow and Fink[1] have shown that

$$\rho = \rho_0(1 + Ap - Bp^2)_t \tag{2}$$

where ρ = density, g per cc at pressure p and temperature t

ρ_0 = density at atmospheric pressure and temperature t

p = pressure, psi

A and B are parameters depending upon temperature t. Curves are given in the reference article showing A and B as functions of t. It may be seen that within the operating temperature encountered in hydraulic work, there is relatively little variation in these factors, so that values at 100°F may be used without introducing an appreciable error. At 100°F, A becomes 4.38×10^{-6} and $B = 5.65 \times 10^{-11}$.

Then

$$C = \frac{A - 2Bp}{1 + Ap - Bp^2} \tag{3}$$

Substitution of values for p in Eq. (3) shows that C varies from 4.25×10^{-6} at 1,000 psi to 3.13×10^{-6} at 10,000 psi. At the most frequently used pressure range, from 1,000 to 3,000 psi, C varies from 4.25×10^{-6} to 3.98×10^{-6}. An average value of 4×10^{-6} for the ordinarily used range of hydraulic pressures seems indicated. The factor C represents the specific change in density per unit of pressure, or the

[1] R. B. Dow and C. E. Fink, Some Properties of Lubricating Oils at High Pressure Density, *J. Applied Phys.* **11**, 353, May, 1940.

change in specific volume, which is proportional to the reciprocal of density. At 3,000 psi, for instance, the total compression is 1.2 per cent of the entire volume under pressure.

How important this property of hydraulic fluid is may readily be seen in the design of hydraulic-pressure machinery for extremely high speed. At every stroke of the unit, the pump must make up the compression volume. If the pump is of constant-delivery type and its discharge is Q in cubic inches per second, then with a given cylinder volume V in cubic inches, the time t required to generate the pressure p in pounds per square inch is

$$t = \frac{Vp}{250{,}000Q} \tag{4}$$

If the discharge of the pump is not constant, but variable, as in many variable-delivery pump applications, a considerably longer time is required for the generation of the pressure. In most variable-delivery controls, the discharge of the pump will decrease with increasing pressure. In the majority of cases, Q is a linear function of pressure p. The pump delivers its full volume up to a given pressure p_1. From there on, the delivery decreases with increasing pressure p until it reaches zero, or better, an infinitely small volume Q_{min} at the maximum operating pressure p_{max}. In the following we assume that p_1 is zero. If it is not zero, of course, Eq. (4) may be used up to p_1, and the variable discharge formula will apply from p_1 to p_{max}. Referring to the diagram (Fig. 1), we have

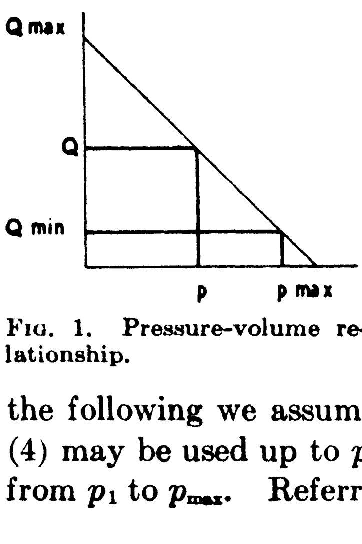

FIG. 1. Pressure-volume relationship.

$$\frac{Q_{max} - Q_{min}}{p_{max}} = \frac{Q_{max} - Q}{p} \tag{5}$$

or

$$Q = Q_{max} - \frac{p}{p_{max}}(Q_{max} - Q_{min}) \tag{6}$$

If $Q = f(p)$, we substitute for Eq. (4)

$$dt = \frac{V\,dp}{250{,}000f(p)} \tag{7}$$

Substituting Eq. (6) for $f(p)$, we have

$$dt = \frac{V\,dp}{250{,}000[Q_{max} - (p/p_{max})(Q_{max} - Q_{min})]} \tag{8}$$

This may be written

$$dt = \frac{V\,dp}{250{,}000Q_{max}} \left\{ \frac{1}{1 - (p/p_{max})[1 - (Q_{min}/Q_{max})]} \right\} \tag{9}$$

or

$$t = \frac{V}{250{,}000Q_{max}} \int_0^{p_{max}} \frac{dp}{1 - p\left[\dfrac{1 - (Q_{min}/Q_{max})}{p_{max}}\right]} \tag{10}$$

Integrating Eq. (10) and substituting limits,

$$t = \left(\frac{V}{250{,}000Q_{max}}\right)\left[\frac{p_{max}}{1 - (Q_{min}/Q_{max})}\right] \log_e \frac{Q_{max}}{Q_{min}} \tag{11}$$

Time t, therefore, is still proportional to the maximum pressure p_{max}, but also to a function of the ratio between the capacities Q_{max} and Q_{min}.

We may compare Eq. (11) with Eq. (4) for time t with constant delivery Q. In this case, Q_{max} would equal Q_{min}. If we substitute Q_{max} for Q_{min} in Eq. (11), we obtain the indeterminate form 0/0.

The actual value of t may be obtained by forming the quotient of derivatives. We write Eq. (11) as follows:

$$t = \frac{Vp_{max}}{250{,}000Q_{max}} \left[\frac{\log_e (Q_{max}/Q_{min})}{1 - (Q_{min}/Q_{max})}\right] \tag{12}$$

Then

$$\frac{f'Q_{min}}{\phi'Q_{min}} = \frac{Q_{max}}{Q_{min}} \tag{13}$$

By substituting Q_{max} for Q_{min} we obtain Eq. (4), which is the constant-delivery formula for the time t.

If we make Q_{min} = zero in Eq. (11), time t will become infinity. Actually this condition will not be realized, as the pump pressure control lags behind the pressure build-up, and as the maximum pressure is built up, the pump is still on a finite discharge stroke Q_{min}.

Eq. (11) will enable us to compute the time required to build up pressure for any given final discharge rate; practical experience seems to indicate a ratio of Q_{max} to Q_{min} of about 10:1. With this figure we arrive at the time

$$\begin{aligned} t &= \left(\frac{V}{250{,}000Q_{max}}\right)\left(\frac{p_{max}}{1 - 0.1}\right) \log_e 10 \\ &= \frac{V}{250{,}000Q_{max}} 2.5p_{max} \end{aligned} \tag{14}$$

or 2½ times the time required to build up pressure with constant discharge.

In actual practice, the pressure p_1 at which the pump discharge will begin to diminish will not be zero, but a fraction of the maximum pressure, generally from 50 to 75 per cent. In this case, Eq. (4) may be used from zero to p_1, and Eq. (11) from p_1 to p_{max}. The total time, therefore, for a variable-delivery pump with constant delivery up to p_1 and decreasing delivery from p_1 to p_{max} would be:

$$t = \frac{V}{250{,}000Q_{max}}\left[p_1 + \frac{p_{max} - p_1}{1 - (Q_{min}/Q_{max})}\log_e \frac{Q_{max}}{Q_{min}}\right] \tag{15}$$

Contrasted with the low compressibility of liquids, gaseous fluids are highly compressible. Owing to the increasing importance played by gas- and air-loaded accumulators and other auxiliary devices, a full understanding of the behavior of gaseous fluids under pressure is desirable, and in the following the fundamental laws governing them will be discussed briefly.

2. Boyle's Law. A quantity of fluid placed in a container will exert a pressure on the walls of the container. This pressure in the case of a liquid is generally produced by the liquid's own weight but may, of course, be increased by the action of additional forces, as by the action of a hydraulic pump or similar means. Gases exert pressure, which is due both to their weight and to molecular action. Fluid pressures are measured in units of force per unit area, such as pounds per square inch or pounds per square foot.

Boyle's law states that the volume of a gas is inversely proportional to the pressure of the gas. Expressed algebraically

$$\frac{p_1}{p} = \frac{v}{v_1} \qquad \text{or} \qquad p_1v_1 = pv = C \tag{16}$$

p and p_1 are absolute pressures and v and v_1 are specific volumes. The law, of course, also applies to absolute volumes.

3. Gay-Lussac's Law. Gay-Lussac's law states that gases under constant pressure expand in proportion to an increase in temperature. All gases have the same coefficient of expansion. Expressed algebraically

$$v_t = v_0(1 + \alpha t) \tag{17}$$

where v_0 = volume of gas at zero temperature
α = coefficient of expansion

If the volume of the gas is kept constant, the pressure increases in proportion to the increase in temperature.

$$p_t = p_0(1 + \alpha t) \tag{18}$$

For $t = -1/\alpha$, p_t becomes zero, and this temperature is known as "absolute zero." $1/\alpha$ is 459.4°F, or 273°C. If temperatures are measured from absolute zero as starting point, they are called "absolute temperatures."

4. Equation of State; The Gas Constant. According to Boyle's law,

$$p_1v_1 = pv = C \quad \text{at constant temperature}$$

At zero temperature,

$$p_0v_0 = pv \tag{19}$$

At temperature t,

$$p_0v_t = pv \tag{20}$$

or, according to Gay-Lussac's law,

$$p_0v_0(1 + \alpha t) = pv \tag{21}$$

Introducing the absolute temperature $T = (1/\alpha) + t$,

$$pv = p_0v_0\alpha T$$

Since v and v_0 are specific volumes, $p_0v_0\alpha$ is a constant for a given gas, the so-called "gas constant," which will be denoted by the symbol R. Then

$$pv = RT \tag{22}$$

The dimension of the gas constant is

$$R = \frac{pv}{T} = \frac{\text{lb}}{\text{sq ft}}\,\frac{\text{cu ft}}{\text{lb}}\,\frac{1}{\text{degree}} = \frac{\text{ft}}{\text{degree}} \tag{23}$$

5. Viscosity. One of the properties of fluid most vital in the operation of hydraulic devices is the viscosity. Viscosity may be defined as the stickiness or resistance to fluid flow of a substance. More accurately, it is the property by virtue of which a fluid offers resistance to a shearing displacement.

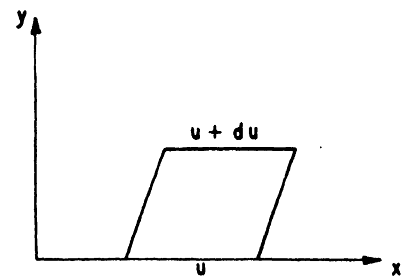

FIG. 2. Displacement of fluid particle.

From this characteristic of fluids a mathematical conception for this property of fluids has been derived. This conception is based on the following fundamental hypothesis, first proposed by Newton: that the shearing stress between adjacent layers of fluid of infinitesimal thickness is proportional to the rate of shear in the direction perpendicular to the motion. If displacement takes place in a particle of fluid owing to fluid flow at a velocity u on its lower surface and $u + du$ on its upper, it will be distorted, and the rate of distortion in the direction perpen-

dicular to the flow may be expressed as $\partial u/\partial y$, the partial derivative of u with respect to y. Newton's law may then be written

$$\tau = \mu \frac{\partial u}{\partial y} \tag{24}$$

μ is a coefficient of proportionality and is called "absolute viscosity." In the case of two parallel plates separated by a distance b, one of which is stationary, while the other moves at a velocity V, the fluid is assumed to adhere at the boundary to each one of the plates, so that its velocity is

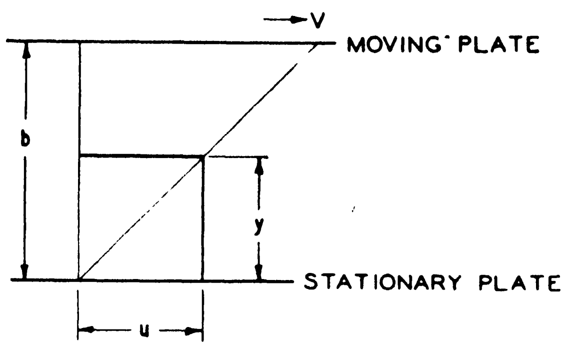

FIG. 3. Parallel plates.

assumed to be zero at the stationary plate and V at the moving plate. Then, at any point y above the lower plate the velocity is

$$u = \frac{Vy}{b} \tag{25}$$

The rate of shear is

$$\frac{\partial u}{\partial y} = \frac{V}{b} \tag{26}$$

The shearing stress is

$$\tau = \mu \frac{V}{b} \tag{27}$$

or

$$\mu = \tau \frac{b}{V} \tag{28}$$

Therefore, the absolute viscosity is a force acting on the unit of area of a plane surface moving at unit velocity relative to another plane area separated from it by a unit distance.

Kinematic Viscosity. Dividing the absolute viscosity μ by the density of the fluid ρ, results in a quantity ν, known as "kinematic viscosity."

$$\nu = \frac{\mu}{\rho} \tag{29}$$

6. Units of Viscosity. The absolute viscosity expressed in units of the English gravitational system is

$$\mu = \frac{\tau b}{V} = \frac{\text{force}}{\text{area}}\frac{\text{length}}{\text{velocity}} = \left(\frac{F}{L^2}\right)\left(\frac{L}{L/T}\right) = \frac{FT}{L^2}$$

Therefore the absolute viscosity may be measured in pound-seconds per square foot in English gravitational units, or dyne-seconds per square centimeter in metric units.

One dyne second per square centimeter is called "one poise," in honor of Poiseuille. One-hundredth of a poise is called "one centipoise." The viscosity of water at 20°C is approximately 1 centipoise.

The kinematic viscosity has the dimension L^2/T or square feet per second in the English and square centimeters per second in the metric system. One square centimeter per second is called "one stoke" in honor of Sir George Stokes. The hundredth part of a stoke is called a "centistoke."

Conversion of English and Metric Units

$$1\,\frac{\text{lb-sec}}{\text{sq ft}} = \frac{453.6 \times 980.665}{30.48^2} = 478.8\,\frac{\text{dyne-sec}}{\text{sq cm}} \text{ or poises}$$
$$= 47{,}880 \text{ centipoises}$$

$$1\,\frac{\text{sq ft}}{\text{sec}} = 30.48^2 = 929.03\,\frac{\text{sq cm}}{\text{sec}} \text{ or stokes} = 92{,}903 \text{ centistokes}$$

7. Viscosimetry. In practice, the determination of the coefficients of absolute and kinematic viscosity is not possible by any direct means. Devices have been developed to measure the speed of flow of viscous fluids through tubular orifices, or the time required for a given quantity of fluid to escape through such an orifice. The measurements thus made bear a definite quantitative relationship to the viscosity as expressed in units of mass, time, and length.

The most widely used method of determining viscosity is the transpiration method. In these instruments the time required for a given quantity of fluid to flow through a capillary tube at a given pressure is determined. The Saybolt viscosimeter is commonly used in the United States for this purpose. It has been adopted by the American Society for Testing Materials as standard. For lubricating and hydraulic oils, the Saybolt universal viscosimeter is employed.

The viscosity is measured in seconds required for 60 cc of liquid to flow vertically through the capillary tube.

8. Conversion of Saybolt Units into Metric Units of Kinematic Viscosity. The relationship between the transpiration time t in Saybolt

seconds and the kinematic viscosity ν in centistokes may be expressed by the following empirical formula:

$$\nu = 0.226t - \frac{195}{t} \text{ centistokes} \qquad \text{for } t < 100 \text{ sec} \tag{30}$$

and

$$\nu = 0.220t - \frac{135}{t} \text{ centistokes} \qquad \text{for } t > 100 \text{ sec} \tag{31}$$

9. Conversion of Saybolt Units into Metric Units of Absolute Viscosity. *a*. Given Saybolt viscosity $= t$.

$$\text{Density} = \rho \qquad \text{g per cc}$$

Then

$$\nu = 0.226t - \frac{195}{t} \text{ centistokes} \qquad \text{for } t < 100 \text{ sec}$$

or

$$\nu = 0.220t - \frac{135}{t} \text{ centistokes} \qquad \text{for } t > 100 \text{ sec}$$

$$\mu = \nu\rho \text{ centipoises} = \frac{\nu\rho}{100} \text{ poises}$$

b. Given Saybolt viscosity $= t$.

Sp. wt. $= w$ g per cc, numerically equal to ρ

Then ν as above in centistokes

$$\mu = \nu w \qquad \text{centipoises}$$

c. Given Saybolt viscosity $= t$.

Sp. gr. $= s$, numerically equal to ρ

ν as above in centistokes

$$\mu = \nu s \qquad \text{centipoises}$$

10. Conversion of Saybolt Units into English Gravitational Units. *a*. Given Saybolt viscosity $= t$.

$$\text{Density} = \rho \qquad \text{slugs per cu ft}$$

Then

$$\nu = \frac{0.226t - (195/t)}{92{,}903} \text{ sq ft per sec} \qquad \text{for } t < 100 \text{ sec}$$

$$= \frac{0.220t - (135/t)}{92{,}903} \text{ sq ft per sec} \qquad \text{for } t > 100 \text{ sec}$$

$$\mu = \nu\rho \qquad \text{lb-sec per sq ft}$$

b. Given Saybolt viscosity = t.

$$\text{Sp. wt.} = w \qquad \text{lb per cu ft}$$

$$\mu = \frac{\nu w}{32.17} \qquad \text{lb-sec per sq ft}$$

c. Given Saybolt viscosity = t.

$$\text{Sp. gr.} = s$$

$$\mu = 1.94\nu s \qquad \text{lb-sec per sq ft}$$

11. Viscosity and the Operation of Hydraulic Machinery. The viscosity of the operating oil is of the utmost importance in the design and operation of oil hydraulic machinery. As will be shown later, hydraulic pumps, valves, and controls for oil-pressure operation are made without internal packings, relying entirely on close fits for creating and maintaining oil pressure. Leakage occurs through the clearances, resulting in loss of capacity and power and increase in temperature. The leakage losses are inversely proportional to the viscosity of the oil. The use of a heavy-bodied oil therefore seems indicated where it is desirable to minimize leakage losses. On the other hand, viscosity produces torque resistance in closely fitted parts, and this resistance is directly proportional to the viscosity, so that for maximum mechanical efficiency lighter oils should be used. Viscosity also influences the flow resistance of oil through valves, pipes, and passages. It increases the difficulties of maintaining proper suction and causes sluggishness in response of control equipment. Oil recommendations of the manufacturers are based on the attempt to reconcile these seemingly incompatible requirements. These recommendations are largely based on experience and tests under actual operating conditions, but attempts have been made at analytical treatment of the problem, which will be dealt with in Chap. V.

The question what viscosity of oil to select cannot be answered unequivocally, as a great many factors influence this decision. Generally speaking, heavy-bodied oils should be used in high-pressure systems, where it is desired to minimize leakage losses caused by high pressure and to prevent excessive thinning out of oil due to high temperature. In a high-pressure system greater torque losses may be tolerated in overcoming viscous shear, and higher pressures may be allowed to drive oil through pipe lines and valves, as the percentage loss in power is not as great as in a low-pressure system. On the other hand, a low-pressure system, say up to 1,000 psi, can be operated to greater advantage with a less viscous oil, as leakage losses tend to be smaller, and loss of power due to viscous shear and pipe-line resistance must be held down.

The author's experience has been that oils from 500 to 950 SSU viscosity at 100°F give good results in radial plunger pumps at pressures

of 2,500 psi and over, while for low-pressure applications at about 1,000 psi, oils of 150 to 300 SSU seem indicated. Several manufacturers of high-pressure pumps recommend oils of 300 SSU viscosity, and the general tendency seems to be toward lighter bodied oils.

12. Temperature and Viscosity. Mineral oils used in hydraulic work are highly sensitive to changes in temperature. Relatively slight changes in temperature produce great changes in viscosity. The relative change

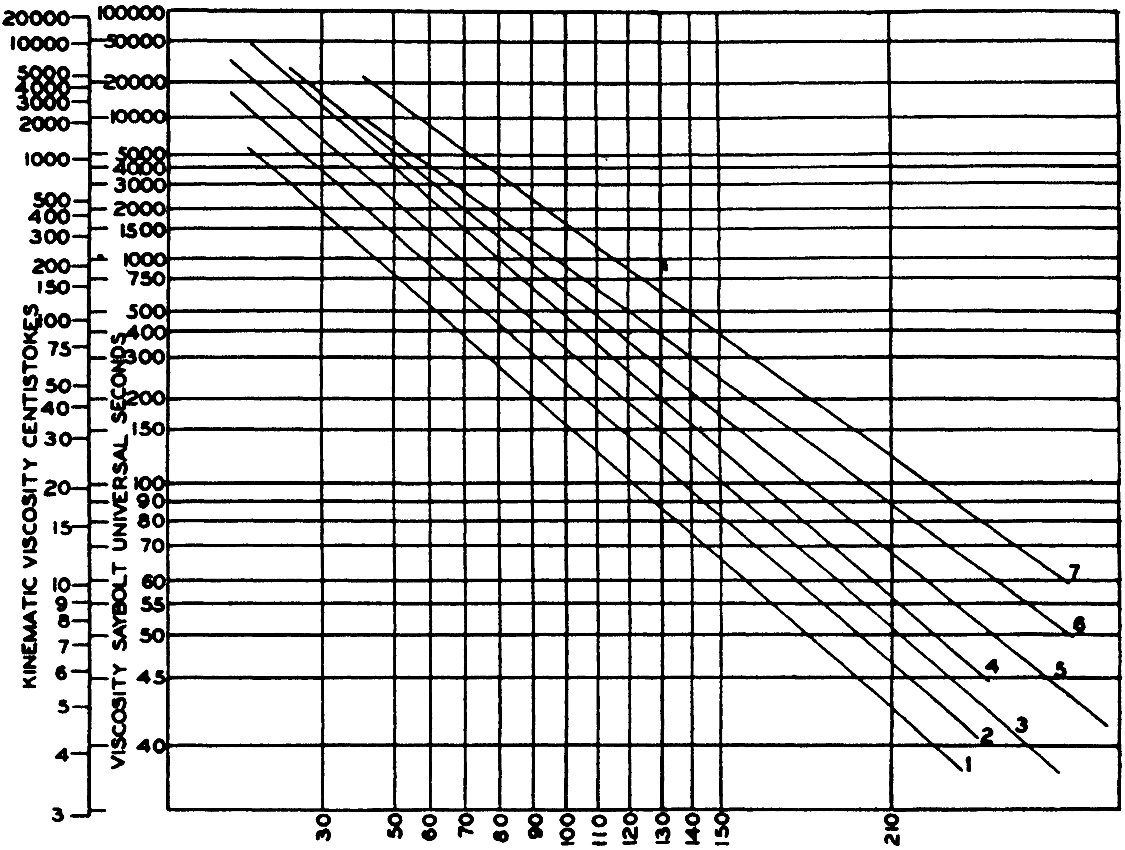

FIG. 4. Viscosity-temperature curves of typical hydraulic oils.

in viscosity per degree Fahrenheit is known as the "temperature coefficient of viscosity" and may be expressed as follows:

$$T_c = \frac{d\mu}{\mu\, dt} \tag{32}$$

The relationship between viscosity and temperature may be expressed by the following empirical equation, upon which the ASTM charts are based.[1]

$$\log \log (\nu + 0.8) = n \log T + C \tag{33}$$

[1] Standard Viscosity-temperature Charts for Liquid Petroleum Products, ASTM Designation D 341-43.

where ν = kinematic viscosity, centistokes

T = absolute temperature

C and n are constant for any given oil.

Figure 4 shows the viscosity-temperature curves of typical hydraulic oils, drawn on an ASTM chart.

A simpler though less rigorous expression, which serves well for analytical purposes, is the Herschel relationship, which expresses the viscosity-temperature function between two viscosities μ and μ_0 existing at the temperatures t and t_0, as follows:

$$\mu = \mu_0 \left(\frac{t_0}{t}\right)^K \tag{34}$$

This relationship holds for only a relatively restricted range of temperatures and should not be extrapolated beyond its range of validity. In the Table I, exponents are given for the oils shown in Fig. 4, covering a temperature range of 70 to 130°F. As a fair approximation, viscosity of oils most commonly used in hydraulic work changes with the third power of the temperature gradient within the normal operating range. With a commonly encountered temperature gradient of about 2:1 between cold start and maximum operating temperature, viscosities vary as 8:1.

TABLE I

Oil	*Exponent K*
1	2.53
2	2.81
3	3.00
4	3.24
5	3.27
6	3.10
7	3.25

13. Pressure and Viscosity. Pressure applied to oil will increase its viscosity. At moderate pressures, this increase is relatively slight, but the viscosity increases rapidly with higher pressures. Viscosity varies with pressure in an exponential function that may be expressed as follows:

$$\mu = \mu_0 e^{\beta p} \tag{35}$$

where β = pressure coefficient of viscosity

μ_0 = viscosity at atmospheric pressure

μ = viscosity at pressure p

The pressure coefficient depends upon the type of oil, temperature, and pressure range.

Table II, showing viscosities at different pressures, has been compiled from data supplied by the Socony Vacuum Oil Co. It may be seen that

the pressure coefficient is not constant, but depends upon the pressure. The changes within the range of pressures used for practical applications are small enough to be neglected, and an average pressure coefficient, assumed to be constant for the entire pressure range, may be used.

TABLE II. EFFECT OF PRESSURE ON THE VISCOSITY OF TYPICAL HYDRAULIC OILS (Temperature = 100°F, viscosities in centipoises, pressures in pounds per square inch)

$p = 14.7$	$p = 1{,}000$		$p = 5{,}000$		$p = 10{,}000$	
μ_0	μ	$\beta \times 10^4$	μ	$\beta \times 10^4$	μ	$\beta \times 10^4$
28.3	33.4	1.65	60.0	1.49	121	1.45
46.4	56.6	1.99	119	1.89	293	1.84
83.1	101	1.99	215	1.90	522	1.84
122	151	2.15	345	2.08	933	2.03
288	351	1.99	714	1.80	1,560	1.69
422	515	1.99	1,050	1.80	2,290	1.69
579	730	2.30	1,630	2.16	4,070	1.96

Second-order Effects. Experiments conducted by Dow[1] indicate second-order effects of pressure on the average temperature coefficient of viscosity and of temperature on the average pressure coefficient. These effects may be neglected within the temperature and pressure range of operating hydraulic equipment, and both coefficients will be assumed to be constant and independent of each other.

14. Requirements for Hydraulic Fluids.[2]

Chemical Stability. In a hydraulic system oil is in constant circulation and is used over and over. Often the oil is churned and agitated and tends to pick up oxygen. Oils that cannot resist this tendency thicken and become sluggish. This slows down the action of the machine and eventually causes it to break down altogether.

Oxidation is accelerated by high temperature. Owing both to this fact and to the rapid loss in viscosity, it is essential to hold down the temperature of the hydraulic fluid, if necessary, by water cooling.

Overheating of an operating oil causes a vicious cycle. Excessive temperatures lower the viscosity; the lowered viscosity in turn increases leakage losses, and the increased leakage losses produce still higher temperatures. This goes on until the excessive temperatures cause a breakdown of the machine, due to seizing and galling caused by destruction

[1] The Effects of Pressure and Temperature on the Viscosity of Lubricating Oils, *J. Applied Phys.* **8**, (No. 5), May, 1937.

[2] Quoted by permission of Socony Vacuum Oil Co., New York, N.Y., from their booklet on hydraulic systems.

of the oil film. The establishment of a proper heat balance is of great importance in a hydraulic system. This heat balance must be based on equal income and outgo of heat under stabilized conditions, and not on any consideration of heat storage in a large reservoir.

Demulsibility. In the operation of hydraulic systems, entrance of moisture into the operating oil cannot always be avoided. Condensation of atmospheric moisture on oil cooler and water pipes, leaky cooler, or leakage of coolant into oil reservoir may be responsible. When water mixes with oil, an emulsion is formed. The oil will turn a yellowish color, become slimy in consistency, or thin out. This emulsion will prove detrimental to the operating parts of the system, cause rusting and rapid wear and deterioration.

Lubricity and Film Strength. It must be kept in mind that hydraulic oils not only serve as power-transmitting mediums, but must also lubricate the moving parts of the pumps, valves, and other parts of the hydraulic equipment. Heavy pressures are developed in the pumps operating at high speeds, and hydraulic oils must provide the lubricating films to resist pressures and prevent metal-to-metal contact.

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CHAPTER III

HYDROSTATICS

1. Pascal's Law. The pressure at any point in a static fluid is the same in all directions. The entire art of hydraulics is built upon this simple and fundamental doctrine.

Pressure and Elevation. Let us consider the equilibrium of forces on a body of fluid in static condition. Figure 5 shows a column of fluid of height h and area A. The forces acting on this column of fluid are the pressures p_1 and p_2 and its weight whA. To satisfy a condition of equilibrium, we must have

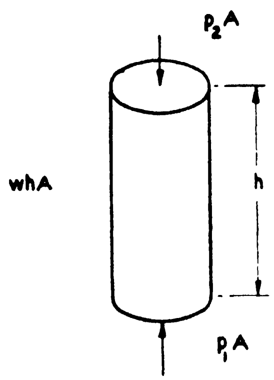

FIG. 5. Forces on fluid column.

$$p_2A + whA = p_1A \tag{1}$$

or

$$p_2 + wh = p_1 \tag{2}$$

$$p_1 - p_2 = wh \tag{3}$$

If p_2 is assumed to be atmospheric pressure on the surface of the fluid, then pressure p at depth h is

$$p = p_a + wh \tag{4}$$

In this equation p is called "absolute pressure," that is, pressure due to the weight of the fluid column plus the weight of the atmosphere above it.

If the atmospheric pressure p_a is assumed to be zero, then we have

$$p = wh \qquad \text{or gauge pressure} \tag{5}$$

This is the pressure that would be shown by a gauge connected at depth h, owing to the fact that the pressure-sensitive element of the gauge is subjected to an atmospheric pressure equal and opposed to that acting on the fluid column, and thus the atmospheric pressure is balanced out.

p_a, the atmospheric pressure, is 14.7 psi at sea level or 2,116 psf. (English atmosphere = 29.9 in. of mercury, or 760 mm. A metric atmosphere is arbitrarily assumed to be 1 kg per sq cm or 736 mm.)

To make Eqs. (4) and (5) dimensionally equal, we must convert units as follows:

$$w \text{ for water} = 62.4 \qquad \text{lb per cu ft}$$

Therefore

$$p = 2{,}116 + 62.4h \qquad \text{psfa} \tag{6}$$

$$p = 62.4h \qquad \text{psfg} \tag{7}$$

or, converting to pounds per square inch,

$$p = 14.7 + \frac{62.4}{144}h = 14.7 + 0.433h \qquad \text{psia} \qquad (8)$$

$$p = 0.433h \qquad \text{psig} \qquad (9)$$

The depth of liquid h to produce a given pressure p from Eq. (5) is

$$h = \frac{p}{w} \qquad (10)$$

and is called the "head" in feet.

For water, if p is in pounds per square foot,

$$h = \frac{p}{62.4} \qquad \text{ft} \qquad (11)$$

or if p is in pounds per square inch,

$$h = 2.32p \qquad \text{ft} \qquad (12)$$

For any other fluid with specific gravity s, if p is in pounds per square inch,

$$h = 2.32\frac{p}{s} \qquad \text{ft} \qquad (13)$$

2. Manometers. The basic principle of Pascal's law has been utilized in devices to indicate the pressure existing in a container filled with liquid under pressure.

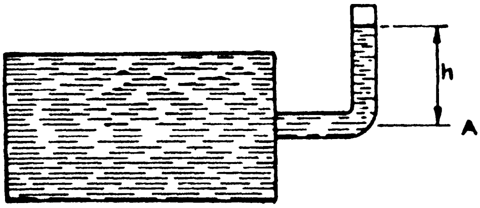

FIG. 6. Piezometer.

If we connect a tube with an open end to a container as shown in Fig. 6, the height of fluid column, h, indicates the pressure prevailing at point A in the container. According to Eq. (4)

$$p = p_a + wh \qquad \text{on the absolute scale}$$

or

$$p = wh \qquad \text{on the gauge scale}$$

The pressure may, of course, be directly read in feet or in inches of head. A tube as shown in Fig. 6 is called a "piezometer tube."

A manometer is a U-shaped tube connected to a vessel under pressure and may indicate gas or liquid pressures. Figure 7 shows a manometer connected to a vessel containing a fluid of a specific weight w_1. The indicating fluid in the tube has the weight w_2. Then

$$p = p_a + w_2h_2 - w_1h_1 \qquad \text{absolute pressure} \qquad (14)$$

or

$$p = w_2h_2 - w_1h_1 \qquad \text{gauge pressure} \qquad (15)$$

Manometers are generally used to measure very low pressures. It may readily be seen that even if an indicating fluid of high specific gravity is used, the amount of pressure that may be measured by this device is subject to practical limitations as to height of manometer tubes. Therefore this type of measuring instrument is not in general use for hydraulic

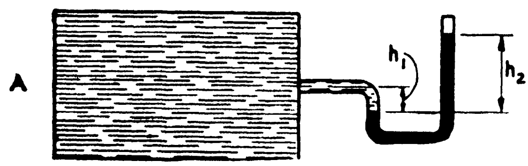

FIG. 7. Manometer.

work except where relatively low pressures must be measured with great accuracy.

3. Pressure Gauges. For the measurement of higher pressures, it is necessary to use another form of pressure gauge. These gauges consist essentially of an elastic pressure-responsive element operating a pointer or dial by means of a mechanism that magnifies the elastic deformation. Most commonly used are Bourdon-type pressure gauges. The pressure-responsive element in this gauge is a curved tube with thin walls that tends to straighten out when subjected to hydraulic pressure (Fig. 8). This movement is transmitted by levers and gear segment to the pointer of the gauge. Dials are graduated in pounds per square inch and may be supplied with an additional graduation showing tons on a ram of given size.

Plunger- and spring-type gauges avoid the complications caused by the transmitting mechanism and have been adopted by many hydraulic manufacturers for their ruggedness and ability to take overload and shock. They are not designed for "pin-point" accuracy of readings, but serve as excellent indicating mediums.

The gauge made by Schrader and Sons in Brooklyn has found considerable use in hydraulic work and is shown in Fig. 9. The gauge consists of a plunger operating against a coil spring, and as the pressure

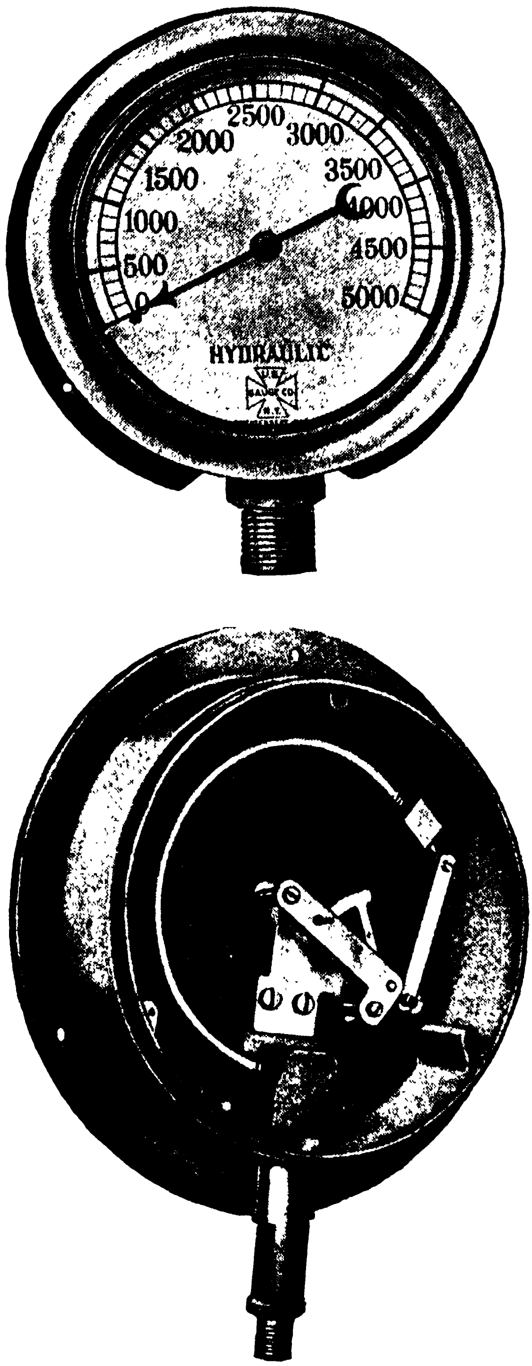

FIG. 8. Bourdon-type pressure gauge. (*U.S. Gauge Co., Sellersville, Pa.*)

increases, the plunger rises, compressing the spring. A pointer is attached to the plunger, indicating the pressure on a calibrated scale.

A gauge has recently been put on the market by the Cameron Iron Works that operates on a novel principle. Originally developed for oil field work, the gauge possesses great ruggedness and appears well adapted to withstand the rigorous service imposed upon it by high-speed hydraulic-press work and similar applications.

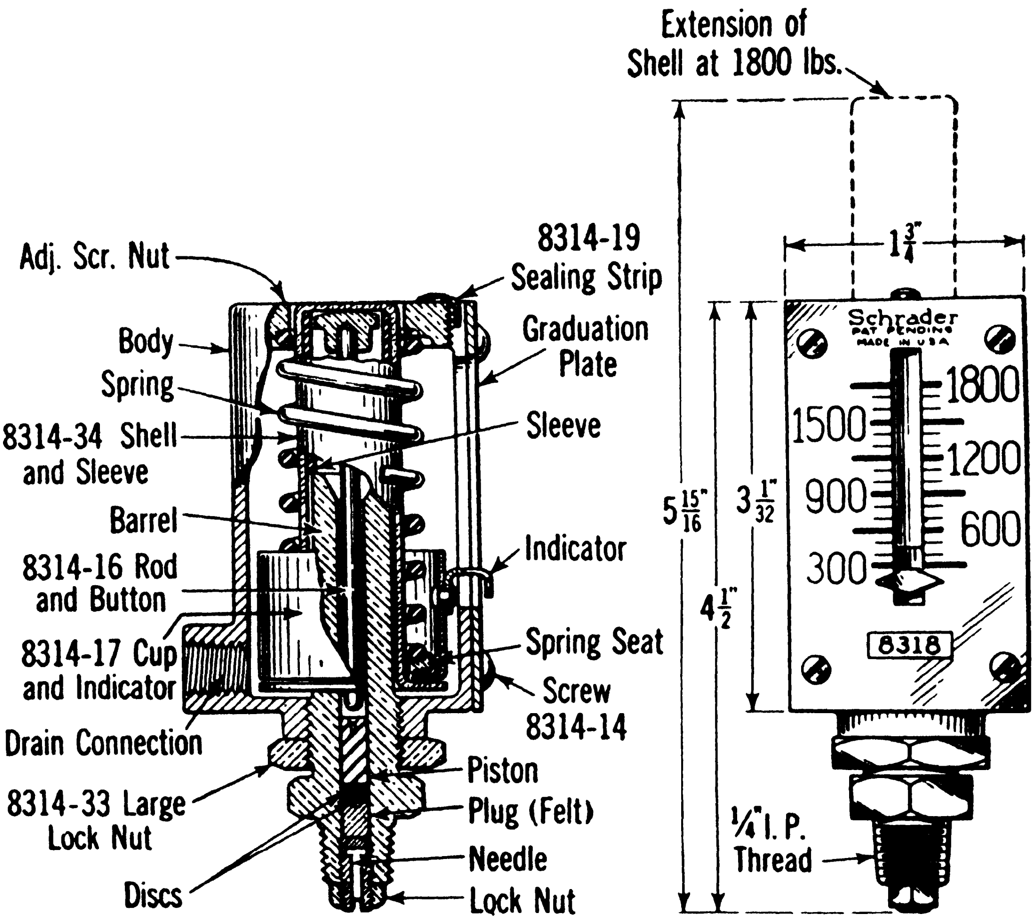

FIG. 9. Schrader pressure gauge. (*A. Schrader and Sons, Brooklyn, N.Y.*)

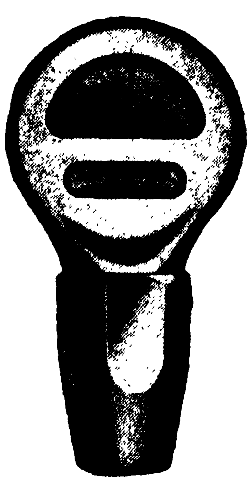

FIG. 10. Cameron pressure gauge. (*Cameron Iron Works, Houston, Texas.*)

CHAPTER IV

HYDRODYNAMICS

1. Continuity. If fluid flows in a pipe or conduit of variable cross section, and assuming that there are no cavities, then the mass of fluid passing any one cross section must be constant.

If the velocities and areas of the conduit on two given cross sections are V_1, A_1, and V_2, A_2, respectively, then

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 \qquad (1)$$

In case of an incompressible fluid, there is no change in density between any two points, and Eq. (1) may be written

$$V_1 A_1 = V_2 A_2 \qquad (2)$$

or, generally,

$$AV = \text{constant} \qquad (3)$$

Equation (3) expresses mathematically the law of continuity of flow.

2. Energy of Fluids. Of the different forms of energy existing, three are of particular importance in dealing with fluid power. They are kinetic, potential, and pressure energy. Energy is measured in foot-pounds in the English gravitational system. When referred to the unit of weight of a fluid, the specific energy is measured in foot-pounds per pound or in feet. Thus liquid stored at a level of 10 ft has a potential energy of 10 ft-lb per lb or 10 ft.

Kinetic energy is similarly measured. The kinetic energy of a mass M at a velocity V equals

$$E = \frac{MV^2}{2} = \frac{WV^2}{2g} \qquad (4)$$

Expressed in energy per unit weight, the energy is

$$E = \frac{V^2}{2g} \qquad \text{ft-lb per lb or feet} \qquad (5)$$

This energy is commonly called the "velocity head."

If a quantity of fluid is subjected to a pressure p, then this condition represents an ability of the fluid to do work. If the total weight of the fluid is W, then the energy at the height h is

$$E = Wh \tag{6}$$

Since

$$h = \frac{p}{w} \qquad \text{[Eq. (10), Chap. III]}$$

$$E = W\frac{p}{w} \tag{7}$$

Again energy per unit weight, p/w, is measured in feet and called "pressure head."

3. Bernoulli's Theorem. If we apply the principle of conservation of energy to the different forms of energy that may occur in a fluid, we have

$$\text{Total energy} = \text{potential} + \text{kinetic} + \text{pressure} \tag{8}$$

$$Wh + \frac{WV^2}{2g} + \frac{Wp}{w} = \text{constant} \tag{9}$$

If expressed in terms of energy per unit weight,

$$h + \frac{V^2}{2g} + \frac{p}{w} = \text{constant} = H \tag{10}$$

H is the total head in feet.

Equation (10) is known as "Bernoulli's theorem" and is named after the mathematician Daniel Bernoulli, who stated this theorem in 1738.

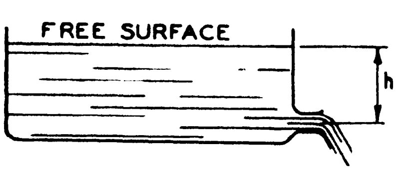

FIG. 11. Toricelli's theorem.

4. Toricelli's Theorem. Assuming that the tank shown in Fig. 11 is filled with fluid to the height h above the outlet opening, then we may apply Bernoulli's law as follows:

At the surface we have a pressure head p_a/w and a potential head of h. The velocity head at this point may be neglected, provided the vessel is of sufficient area to cause a relatively slow movement of fluid.

At the outlet nozzle we have a velocity head $V^2/2g$ and a pressure head p_a/w.

Then

$$\frac{p_a}{w} + h = \frac{p_a}{w} + \frac{V^2}{2g} \tag{11}$$

or

$$h = \frac{V^2}{2g} \qquad \text{or} \qquad V = \sqrt{2gh} \tag{12}$$

This corresponds to the velocity of a body falling in a vacuum a distance h.

Equation (12) is known as "Toricelli's theorem" and was demonstrated by Toricelli in 1644.

5. The Siphon. Bernoulli's theorem may be used to analyze the action of a siphon. Figure 12 shows a single diagram of the action of a siphon. Fluid in the tank flows through the U-shaped tube and out at the bottom, after the siphon has been "primed," that is, after the air has been removed from the tube.

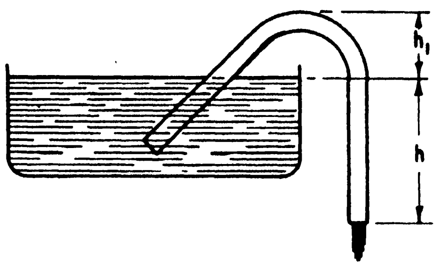

FIG. 12. The siphon.

Applying Bernoulli's law to the level of fluid in the tank and the outlet opening, we have

$$\frac{p_a}{w} + h = \frac{p_a}{w} + \frac{V^2}{2g}$$

or

$$V = \sqrt{2gh}$$

the same as in the case of outflow from a container with head h.

Applying Bernoulli's law to the summit of the siphon tube and the fluid level, we have

$$\frac{p_a}{w} + h = \frac{V^2}{2g} + h + h_1 + \frac{p}{w} \tag{13}$$

To maintain continuity, V must remain constant, assuming the siphon tube is of constant cross section throughout.

Then

$$V = \sqrt{2gh} \qquad \text{or} \qquad V^2 = 2gh$$

$$\frac{p_a}{w} + h = \frac{2gh}{2g} + h + h_1 + \frac{p}{w} \tag{14}$$

or

$$\frac{p}{w} = \frac{p_a}{w} - (h + h_1) \tag{15}$$

when $h + h_1$ equals p_a/w, then the absolute pressure p becomes zero and the siphon ceases to function. The atmospheric pressure p_a is unable to maintain the flow by the combination of heads h and h_1.

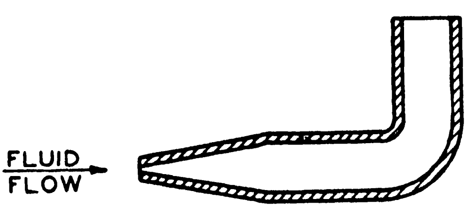

FIG. 13. The Pitot tube.

6. The Pitot Tube. Another well-known application of Bernoulli's theorem is the Pitot tube, named after the French scientist who was the first to use it. Pitot tubes are used to measure the velocity of fluid flow.

The Pitot tube is a tube inserted into the fluid stream with one leg pointing up. The end submerged in the stream is open, while the vertical leg is connected to a manometer. The fluid in the tube is stationary; therefore V at the mouth must be zero.

Applying Bernoulli's theorem to the mouth of the tube, we have

$$\frac{p_m}{w} = \frac{V^2}{2g} + \frac{p}{w} \tag{16}$$

where p_m = pressure at mouth
p = static pressure in fluid

$$w = \rho g$$

$$p_m = \frac{\rho V^2}{2} + p$$

$$V = \sqrt{\frac{2(p_m - p)}{\rho}} \tag{17}$$

The determination of the static pressure p in the stream of fluid requires special devices. Pressure immediately adjacent to the wall may be measured by means of a piezometer. Pressures in the path of flowing fluid are measured by means of so-called "static tubes" (Fig. 14). A static tube consists of a horizontal section of tube closed at the end and fitted with a tapered nose, so as to minimize the disturbance of fluid flow. A number of small holes are drilled a suitable distance away from the nose of the horizontal section. The vertical leg is connected to a manometer. Since fluid flow passes at right angles to the axes of the small holes, only static pressure is transmitted to the manometer.

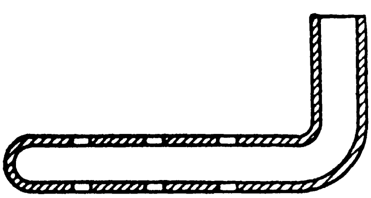

FIG. 14. Static tube.

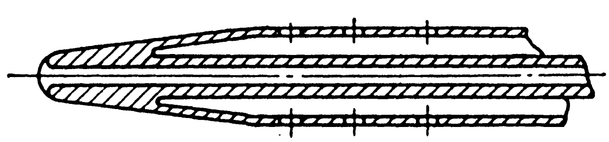

FIG. 15. Pitot-static tube.

Pitot and static tubes may be combined conveniently into Pitot-static tubes as shown in Fig. 15. The two passages in a Pitot-static tube may be connected to opposite ends of a differential manometer, which will then indicate the velocity head.

Equation (17) in actual practice must be corrected by a calibration coefficient to care for interference effects due to the presence of the tube in the stream.

Pitot-static tubes, while primarily intended to measure velocity of flow, may, of course, be used to measure quantity of fluid flowing. If A is the area of pipe or conduit in which the Pitot-static tube is placed, in square feet,

$$Q = AV \qquad \text{cu ft per sec} \tag{18}$$

Calibration coefficients are again necessary to care for variation in velocity in the cross section of the conduit.

7. Venturi Meter. Venturi meters are widely used for measuring quantity of flow in pipes. The device consists of a large section, generally equal to the diameter of the pipe, which tapers down to a neck of smaller diameter, and then a gradually diverging section bringing the area back to that of the pipe (Fig. 16).

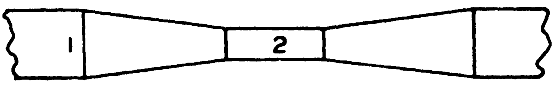

FIG. 16. The Venturi meter.

Establishing Bernoulli's equation for section 1 and 2, we have

$$\frac{p_1}{w} + \frac{V_1^2}{2g} = \frac{p_2}{w} + \frac{V_2^2}{2g} \tag{19}$$

Since

$$A_1V_1 = A_2V_2$$

$$V_2 = \frac{A_1}{A_2} V_1$$

Substituting this value of V_2 in Eq. (19), we have

$$V_1 = \sqrt{2g \frac{1}{(A_1/A_2)^2 - 1}\left(\frac{p_1}{w} - \frac{p_2}{w}\right)} \tag{20}$$

or, if

$$\sqrt{\frac{1}{(A_1/A_2)^2 - 1}} = K$$

$$V_1 = K\sqrt{2g\left(\frac{p_1}{w} - \frac{p_2}{w}\right)} \tag{21}$$

The discharge Q flowing through the meter in cubic feet per minute is

$$Q = A_1V_1 = A_1K\sqrt{2g\left(\frac{p_1}{w} - \frac{p_2}{w}\right)} \tag{22}$$

The figures of Eq. (22) must be modified by a calibration coefficient to allow for the loss of energy and variation in velocity throughout the cross section.

Pressures p_1 and p_2 may be measured by suitably attached gauges or manometers at the entrance and throat sections. These gauges may be graduated to read in terms of discharge or velocity directly.

8. Orifice Meters. The theory of the orifice meter may be developed in a manner similar to that of a Venturi meter. An orifice meter is formed by inserting an orifice in the line where it is desired to measure the flow of fluid and installing manometers or gauges slightly ahead of and behind the orifice. Figure 17 shows the principle of an orifice meter.

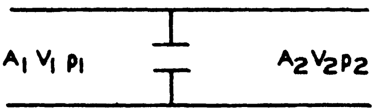

FIG. 17. Orifice meter.

Applying Bernoulli's law, we have

$$\frac{p_1}{w} + \frac{V_1^2}{2g} = \frac{p_2}{w} + \frac{V_2^2}{2g} \tag{23}$$

$$V_1 = \frac{A_2}{A_1} V_2$$

$$\frac{p_1}{w} + \left(\frac{A_2}{A_1}\right)^2 \frac{V_2^2}{2g} = \frac{p_2}{w} + \frac{V_2^2}{2g}$$

$$V_2 = \sqrt{\frac{2g[(p_1/w) - (p_2/w)]}{1 - (A_2/A_1)^2}} \tag{24}$$

$$Q = A_2 \sqrt{\frac{2g\,[(p_1/w) - (p_2/w)]}{1 - (A_2/A_1)^2}} \tag{25}$$

Again a suitable calibration coefficient must be applied in Eq. (25) to allow for the losses of energy and contraction in the orifice.

9. The Variable-opening Flowmeter. The Venturi meter has a constant area of flow, and changes in rate of flow cause velocity changes through this area. These velocity changes cause changes in pressure drop, which are proportional to the square of the velocity change. Instead of maintaining the area of flow constant and changing velocities and pressure heads, it is, of course, possible to vary the opening and maintain a constant head of pressure and constant velocity of flow. An instrument of this type is shown in Fig. 18.

The device, made by Fischer & Porter Co., Hatboro, Pa., consists in principle of a tapered glass tube in which a float of constant cross section is so arranged that it may travel in vertical direction, thus varying the annular area between the float and the inside area of the tube. The upward- and downward-acting forces on the float are in equilibrium, so that the float assumes a definite elevation at a given flow rate.

The downward force is the weight of the float less the weight of the displaced fluid. This must be counterbalanced by a fluid pressure below the float that is greater than the pressure above. Since the weight of the

float is constant at all points, the pressure drop across and the velocity through the annular opening must also be constant. Increased flow rates, therefore, will cause the float to assume a position in the tapered tube corresponding to a larger flow area.

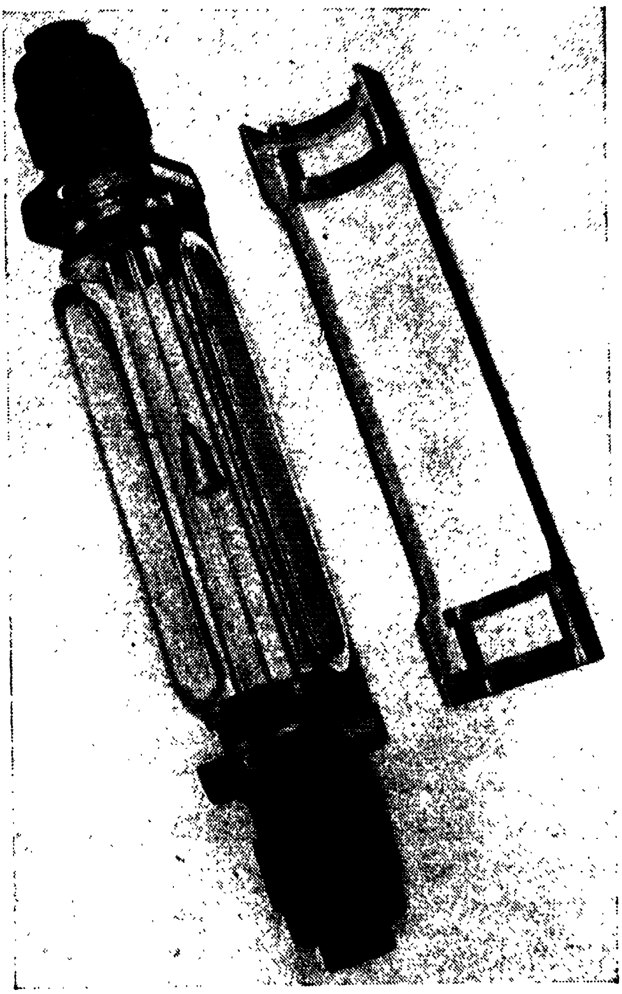

FIG. 18. Flowrator. (*Fischer & Porter Co.*)

Briefly, the theory of the Fischer & Porter flowrator is as follows: The downward-acting force may be expressed as

$$V_f(w_f - w_\phi)$$

where V_f = volume of float
w_f and w_ϕ = specific weights of float and fluid, respectively
The upward-acting force is

$$(p_1 - p_2)A_f$$

where A_f = cross section of float at its widest part
p_1 and p_2 = pressures below and above the float

Equating both expressions, we have

$$A_f(p_1 - p_2) = V_f(w_f - w_\phi) \tag{26}$$

or, if expressed as difference in head,

$$h = \frac{p_1 - p_2}{w_\phi} = \frac{V_f(w_f - w_\phi)}{A_f w_\phi} \tag{27}$$

Applying Toricelli's theorem, we have

$$V = c\sqrt{2gh}$$

in which c is a suitable calibration coefficient. Substituting h from Eq. (27), we have

$$V = \frac{Q}{A} = c\sqrt{\frac{2gV_f(w_f - w_\phi)}{A_f w_\phi}} \tag{28}$$

or

$$Q = Ac\sqrt{\frac{2gV_f(w_f - w_\phi)}{A_f w_\phi}} \tag{29}$$

From Eq. (29) it becomes obvious that flow rates are proportional to areas A and to a function of float and fluid densities, instead of to the square root of a head differential. Extensive tests by the manufacturers of the device have shown that this statement is correct indeed, and that no coefficients other than orifice coefficient c are necessary to cover full range of fluid and float densities. Moreover, by a special design of float developed by considerable research, the influence of viscosity has been largely eliminated, making the coefficient c constant for a wide range of viscosities and densities. Compensation for variation in fluid densities may be made by suitably chosen float densities. For a more complete discussion of the theory of this instrument, the reader is referred to the publications of the maker.

10. Positive-displacement Meters. While positive-displacement meters do not form an application of hydrodynamics in the sense that orifice meters do, a discussion of their basic principle in this chapter is desirable, nonetheless, to familiarize the reader with all the means used to measure and record fluid flow.

It should be kept well in mind that the instruments discussed in Secs. 7 and 8 indicate rates of flow, that is, quantity per unit time, in cubic feet per second, gallons per minute, etc. Contrasted with this, a positive-displacement meter measures total flow in cubic feet or gallons, and to determine rates of flow, time measurements must be taken to

compute the total flow for a given time interval and thus determine rates of flow.

Figure 19 shows a positive-displacement meter of the nutating-plate type. It consists basically of a circular plate mounted in a spherical housing, which is separated into two sections by a radial wall. The plate is slotted to pass this wall. The plate is carried in a spherical bearing and carries an actuating pin, which operates a rotating cam. The plate nutates under the influence of the fluid passing through the meter, and this nutating movement causes rotation of the cam. The rotation is transmitted through a set of gears to the indicating and recording mechanism contained in the housing on top of the meter.

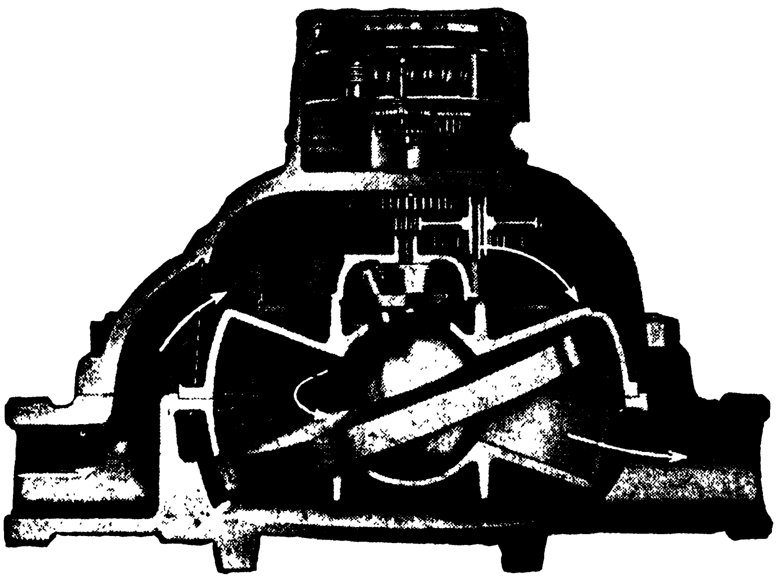

FIG. 19. Positive-displacement meter. (*Buffalo Meter Co.*)

11. Cavitation. When a condition arises in fluid flow where the fluid is not entirely filling out the space provided for it, cavities will form, and this phenomenon is called "cavitation."

Cavitation will occur when the velocity at some point reaches a value so high as to cause the absolute pressure to approach zero. If the suction lines of pumps are of insufficient size, or if obstructions are present that tend to reduce the available pressure head, the pressure may drop so low as to be unable to maintain the required flow, and cavitation will result.

Cavitation will set in when the pressure has dropped to the vapor tension of the fluid at the given temperature, causing the fluid to boil and fill the cavities with vapor. Vapor tension of a fluid is that pressure at which, at a given temperature, free evaporation of the fluid will take place, filling an enclosed space to the saturation point. The vapor tension of hydraulic oils at ordinary operating temperatures is so low that in most cases severe air infiltration will occur, unless the system is

absolutely airtight. As the pressure in the fluid is lowered, any air dissolved in it will tend to liberate, increasing the tendency to cavitate. Entrained air will expand and increase the percentage of cavitation.

Cavitation in hydraulic equipment causes excessive noise in pumps, vibration in pipe lines, and erratic operation of motor and control equipment. It must be avoided by proper design of pumps and passages; suction lines must be of ample size and without restrictions, sharp turns, and sudden changes in cross section.

12. Energy Losses in Fluids. The discussions in the preceding sections of this chapter all referred to ideal fluids, that is to fluids incompressible and without friction. Actually, fluid flow is accompanied by friction due to viscosity and losses due to bends, valves, and obstructions. These energy losses are converted into heat and cannot be recovered. Losses are incurred both in flow in pipes and conduits and in discharge from nozzles and orifices. A detailed study of the losses in fluid flow will be given in the following chapter.

CHAPTER V

VISCOUS FLOW

1. Laminar and Turbulent Flow. The classical experiments conducted by Osborne Reynolds showed that there are two types of flow, which Reynolds called "streamline" and "sinuous" flow.

Reynolds' apparatus consisted of a glass tube inserted into a reservoir filled with water. A valve was placed in the inlet of the tube to control the rate of flow. An auxiliary reservoir containing a colored liquid was arranged above the water tank and fitted with a tube and nozzle discharging the dye into the glass tube at the same velocity as the water flowing in it.

As long as the velocity in the tube was maintained at a sufficiently low value, the jet of dye traveled down the pipe as a straight line. This is what Reynolds called "direct" or "streamline" flow.

Increasing the speed in the tube finally led to the break-up of the jet of colored liquid, which mixed with the surrounding water, indicating what Reynolds called "sinuous" flow.

Streamline flow is also known as "laminar" or "viscous" flow, as contrasted with "sinuous" or "turbulent" flow. It will be shown later that there is a definite criterion that determines whether flow is laminar or turbulent. In the following, we shall deal with the phenomena encountered in viscous or laminar flow.

2. Viscous Flow. In Sec. 5, Chap. II, a fundamental definition of viscosity has been given, and the basic units of absolute and kinematic viscosity have been established.

We have seen that the absolute viscosity μ in the metric absolute system is measured in dyne-seconds per square centimeter or poises, and in the English gravitational system in pound-seconds per square foot. The kinematic viscosity

$$\nu = \frac{\mu}{\rho}$$

is measured in square centimeters per second or stokes, or in square feet per second.

Viscous flow is characterized by the movement of fluid in streamlines or laminae without appreciable turbulence.

3. Viscous Flow in Circular Pipes; Hagen-Poiseuille Law. The law governing viscous flow in circular pipes was based on experiments performed independently by Hagen, a German engineer, and Poiseuille, a French scientist. The law states that the quantity of fluid that flows through a small tube in a given time, is proportional to the pressure difference, to the fourth power of the diameter of the tube, and inversely to its length. A theoretical derivation of this law will be given in the following.

Figure 20 shows a horizontal and vertical cross section of a cylindrical pipe of internal diameter D. If we consider a cylindrical section of length L and diameter $2y$, and assume the fluid moving from left to right

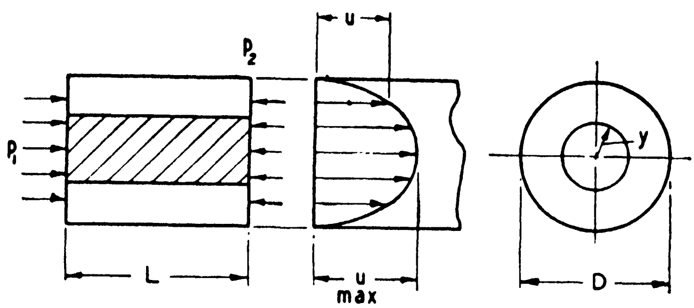

FIG. 20. Viscous flow in circular pipe.

under an average pressure p_1 on the left and p_2 on the right, then the force produced on this cylinder by the pressure difference is

$$(p_1 - p_2)\pi y^2$$

We assume that the flow is steady and that no accelerating force acts on any of the fluid particles. There is, however, a force acting on the outside surface of the cylinder of fluid, due to the shearing stress set up by the viscosity of the fluid. This shearing stress $\tau = \mu\, du/dy$ is independent of x, since there is no change in velocity in the x axis. The shearing force, therefore, is

$$-\mu \frac{du}{dy} 2\pi y L$$

The minus sign indicates that u decreases with increasing y.

To maintain equilibrium of forces, we must have

$$(p_1 - p_2)\pi y^2 = -\mu \frac{du}{dy} 2\pi L y \tag{1}$$

or

$$\frac{du}{dy} = -\frac{(p_1 - p_2)y}{2\mu L} \tag{2}$$

Integration furnishes

$$u = -\frac{(p_1 - p_2)y^2}{4\mu L} + C \tag{3}$$

At the wall of the pipe the fluid is at rest; therefore, for $y = D/2$, u must equal zero, and

$$C = \frac{(p_1 - p_2)D^2/4}{4\mu L}$$

and

$$u = \frac{(p_1 - p_2)}{4\mu L}\left(\frac{D^2}{4} - y^2\right) \tag{4}$$

The maximum velocity at the center of the pipe for $y = 0$ is

$$u_{max} = \frac{(p_1 - p_2)D^2}{16\mu L} \tag{5}$$

The average velocity may be computed by determining the volume of the paraboloid formed by the velocity distribution in the pipe. The average velocity is

$$V = \frac{\text{volume of paraboloid}}{\text{area of base}}$$

$$\text{Volume} = \int_0^{u_{max}} \pi y^2\,du$$

From Eq. (4)

$$u = \frac{(p_1 - p_2)}{4\mu L}\left(\frac{D^2}{4} - y^2\right)$$

If we make

$$\frac{p_1 - p_2}{4\mu L} = C'$$

then

$$y^2 = \frac{D^2}{4} - \frac{u}{C'}$$

$$\int_0^{u_{max}} \pi y^2\,du = \pi u_{max}\frac{D^2}{4} - \frac{\pi u_{max}^2}{2C'}$$

Division of this expression by $\pi D^2/4$ furnishes

$$V = u_{max} - \frac{2u_{max}^2}{C'D^2}$$

or with

$$u_{max} = C'\frac{D^2}{4} \qquad \text{[from Eq. (5)]}$$

$$V = \frac{C'D^2}{8} = \frac{p_1 - p_2}{32\mu L}D^2 \tag{6}$$

The total quantity of fluid passing any cross section in unit time is the product of the average velocity and the area, or

$$Q = \frac{\pi D^4(p_1 - p_2)}{128\mu L} \tag{7}$$

Equation (7) is the mathematical expression of the Hagen-Poiseuille law given at the beginning of this section. This law has been well established by both experimentation and analysis. It serves as a verification of Newton's theory that the shearing stress in a viscous fluid is directly proportional to the velocity gradient in the direction perpendicular to the motion.

Equation (7) is expressed in fundamental units of the English gravitational system. For convenience of calculation, the conversion to more commonly used units will be given in the following.

Given $p_1 - p_2$ in pounds per square inch.

$$\mu = \rho\nu = \frac{\nu w}{g} = \frac{\nu \times 62.4s}{g} \quad \text{with } s = 0.92 \text{ for oil}$$

With ν in centistokes, L in feet, D in inches, Q in cubic inches per minute,

$$\begin{aligned} Q &= \frac{1{,}728 \times 60 \times \pi \times 12^2 \times 92{,}903}{12^4 \times 128 \times 1.78}\left[\frac{(p_1 - p_2)D^4}{\nu L}\right] \\ &= \frac{921{,}850(p_1 - p_2)D^4}{\nu L} \quad \text{cu in. per min} \end{aligned} \tag{8}$$

From this we may compute loss of pressure in a pipe of length L in feet

$$p_1 - p_2 = \frac{Q\nu L}{921{,}850D^4} \tag{9}$$

4. Viscous Flow between Parallel Plates. Figure 21 shows the cross section of flow between two parallel plates taken in the direction of

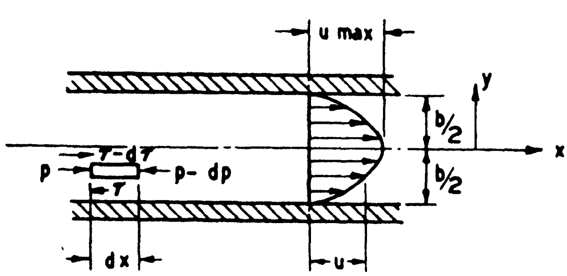

FIG. 21. Viscous flow between parallel plates.

motion. The plates are assumed to be so large that the flow may be considered two-dimensional.

The velocity distribution is assumed to be parabolic; that is, the velocity increases beginning from the walls toward the center, but the rate of velocity increase or the velocity gradient decreases toward center.

The fluid flow is produced by a pressure differential dp and opposed by the shear stress τ acting on the lower surface dx. On the upper surface a shear stress acts in the opposite direction, owing to the fact that the layer of fluid adjacent to the upper surface travels at a faster rate than the fluid particle. This shear stress is smaller than the shear stress on the lower surface, owing to the decrease in the velocity gradient, by the infinitesimal $d\tau$.

To establish equilibrium of forces, we have

$$dp\,dy + (\tau - d\tau)dx = \tau\,dx$$

or

$$dp\,dy - d\tau\,dx = 0$$

Therefore

$$\frac{dp}{dx} = \frac{d\tau}{dy}$$

With

$$\tau = \mu \frac{du}{dy}$$

we have

$$\frac{dp}{dx} = \mu \frac{d^2u}{dy^2} \tag{10}$$

Integrating twice, we obtain

$$u = \frac{1}{\mu}\frac{dp}{dx}\frac{y^2}{2} + A_1 y + A_2 \tag{11}$$

The constants of integration, A_1 and A_2, are determined by the condition that the velocity is zero at the boundaries; that is, for $y = \pm(b/2)$, A_1 becomes zero and

$$A_2 = -\frac{1}{\mu}\frac{dp}{dx}\frac{b^2}{8}$$

Then

$$u = -\frac{1}{2\mu}\frac{dp}{dx}\left(\frac{b^2}{4} - y^2\right) \tag{12}$$

u_{max} occurs in the middle of the space between the surfaces for $y = 0$.

$$u_{max} = -\frac{1}{2\mu}\frac{dp}{dx}\frac{b^2}{4} \tag{13}$$

The velocity distribution is parabolic. The average velocity is

$$V = \frac{\int_0^{u_{max}} y\,du}{b/2} \qquad du = \frac{1}{\mu}\frac{dp}{dx}\,y\,dy$$

$$V = \frac{-(1/\mu)(dp/dx)\int_{b/2}^{0} y^2\,dy}{b/2} = -\frac{1}{3\mu}\frac{dp}{dx}\frac{b^2}{4} \tag{14}$$

The pressure gradient $(dp/dx) = -(p_1 - p_2/L)$, since the pressure decreases in a linear function. Therefore

$$V = \frac{1}{3\mu}\frac{p_1 - p_2}{L}\frac{b^2}{4}$$

The quantity of fluid passing through a unit thickness normal to the xy plane, therefore, is

$$Q = \frac{1}{3\mu}\frac{p_1 - p_2}{L}\frac{b^3}{4} = \frac{(p_1 - p_2)b^3}{12\mu L} \tag{15}$$

For a total thickness w the amount of fluid is

$$Q = \frac{(p_1 - p_2)b^3w}{12\mu L} \tag{16}$$

If we again express Q in Eq. (16) in more convenient dimensional units, we have

$$Q = \frac{3{,}131{,}561(p_1 - p_2)b^3w}{\nu L} \text{ cu in. per min} \tag{17}$$

where ν = kinematic viscosity, centistokes
L = length, ft
w = width, in.
b = thickness, in.
Q = quantity, cu in. per min
$p_1 - p_2$ = pressure gradient, psi

5. Viscous Flow through Annular Spaces. For the flow of fluid through a narrow annular space as shown in Fig. 22, $w = \pi D$. Therefore we have

$$Q = \frac{9{,}833{,}102(p_1 - p_2)b^3D}{\nu L} \tag{18}$$

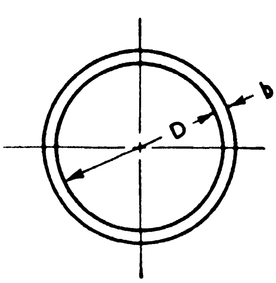

FIG. 22. Viscous flow through annular space.

Equation (18) is of great importance in permitting us to calculate the leakage loss in closely fitted pistons of hydraulic devices. It indicates that the leakage losses increase with the third power of the clearance, which illustrates the necessity of maintaining close clearances and accurate tolerances in hydraulic work.

Clearances can be reduced to very close limits by lapping mating parts together. For interchangeable work, where reasonable manufacturing tolerances must be maintained, inevitably larger clearances will result, so that a moderate slippage or leakage loss will occur in commercially made

hydraulic devices. The clearances and tolerances in Table I have been found satisfactory for commercial work from the author's experience.

TABLE I

Diam. up to, in.	Bore tolerance, in. $\times 10^{-4}$	Piston tolerance, in. $\times 10^{-4}$	Min. clearance, in. $\times 10^{-4}$	Max. clearance, in. $\times 10^{-4}$
$\frac{1}{4}$	$+2\frac{1}{2}-0$	$+0-1\frac{1}{2}$	1	5
$\frac{1}{2}$	$+3-0$	$+0-2$	2	7
$\frac{3}{4}$	$+4-0$	$+0-2\frac{1}{2}$	3	$9\frac{1}{2}$
1	$+5-0$	$+0-3$	5	13
2	$+6-0$	$+0-4$	8	18
3	$+8-0$	$+0-5$	10	23
4	$+8-0$	$+0-5$	$12\frac{1}{2}$	$25\frac{1}{2}$
6	$+10-0$	$+0-6$	17	33
8	$+12-0$	$+0-8$	20	40

The minimum clearances are recommended for satisfactory operation of reciprocating or rotating parts. The maximum clearances result from the addition of both tolerances to the minimum clearances. Minimum clearances may be either added to the nominal bore sizes or subtracted from the nominal shaft sizes, whichever is more convenient.

Example: For $\frac{3}{4}$-in. size

$$\text{Bore} = 0.750 + 0.0003 + 0.0004 = 0.7507 \text{ in.}$$
$$- 0.0000 = 0.7503 \text{ in.}$$
$$\text{Piston} = 0.750 - 0.00025 = 0.74975 \text{ in.}$$
$$+ 0.00000 = 0.75000 \text{ in.}$$

or, alternately,

$$\text{Bore} = 0.750 + 0.0004 = 0.7504 \text{ in.}$$
$$- 0.0000 = 0.7500 \text{ in.}$$
$$\text{Piston} = 0.750 - 0.0003 - 0.00025 = 0.74945 \text{ in.}$$
$$+ 0.00000 = 0.74970 \text{ in.}$$

With the maximum clearance of 0.00095 (diametral) and assuming that the $\frac{3}{4}$-in. piston is fitted with a length of $\frac{1}{2}$ in., we may expect a leakage of about 1 cu in. per min with an oil of 40 centistokes viscosity at a pressure of 2,000 psi and atmospheric terminal pressure.

6. Viscous Flow with Variable Viscosity. Equations for viscous flow given in preceding sections are based on the assumption that the viscosity in the flow path is constant. Actually, this is far from being the case. We have seen in Chap. II that viscosity of oils is highly sensitive to changes in temperature and also susceptible to pressure changes. Therefore, pressure drop through capillary seals and other viscous flow paths is not constant and must be expressed as a differential equation with variable viscosity. The kinematic viscosity used in these flow equations

is also influenced by the compressibility due to changes in density. These changes, however, are too slight to be considered in this analysis.

Any of Eqs. (8), (17), or (18) may be expressed in the following simplified form:

$$Q = \frac{C_1\,\Delta p}{\nu\,\Delta L} \tag{19}$$

The coefficient C_1 covers the particular physical configuration of the flow path with which we are dealing.

From Eq. (34), Chap. II, we have

$$\mu = \mu_0\left(\frac{t_0}{t}\right)^l$$

Also from Eq. (35), Chap. II,

$$\mu = \mu_0 e^{\beta p}$$

We may combine Eqs. (34) and (35) by stipulating that μ_0 is viscosity at atmospheric pressure (zero gauge pressure) and existing operating temperature t_0, while μ is the viscosity in the capillary seal at the pressure p and temperature t due to absorption of the pressure energy by the fluid. The temperature rise must be proportional to the pressure energy absorbed. We may write, therefore,

$$t - t_0 = C_2(p_1 - p) \tag{20}$$

where p_1 is the pressure at the seal entrance.

Coefficient C_2 may be computed in the following manner. When a quantity of oil Q is forced through the opening, energy is imparted to it, which results in the absorption of an equal amount of heat energy. If q is the specific heat, we have

$$144Q\,\Delta p = Qwq(t - t_0)778 \tag{21}$$

If we take

$$w = 57.4 \text{ lb per cu ft}$$
$$q = 0.45 \text{ Btu per lb per °F}$$

then from Eq. (21)

$$t - t_0 = (7.2 \times 10^{-3})\Delta p$$

or, from Eq. (20),

$$C_2 = 7.2 \times 10^{-3}(\text{°F})(\text{sq in.}) \text{ per lb}$$

From Eq. (20) we obtain

$$\frac{t}{t_0} = \frac{C_2}{t_0}(p_1 - p) + 1 \tag{22}$$

then

$$\mu = \frac{\mu_0}{[(C_2/t_0)(p_1 - p) + 1]^K} \tag{23}$$

Before proceeding with the analysis a somewhat simplified, if less rigorous expression for μ as temperature function of pressure p will be developed. We have seen that the exponent K can be approximated by the integer 3 for average operating conditions. Moreover, within the pressure and temperature range of practical hydraulic application, the expression $(C_2/t_0)(p_1 - p)$ is small compared with 1, and the expression $[(C_2/t_0)(p_1 - p) + 1]^3$ may be written $3(C_2/t_0)(p_1 - p) + 1$ with an error not exceeding $8\frac{1}{2}$ per cent for $t_0 = 100°$ and $p_1 = 3{,}000$ psi. Then we have

$$\mu = \frac{\mu_0}{3(C_2/t_0)(p_1 - p) + 1} \tag{24}$$

Combining Eq. (24) and Eq. (35), Chap. II, as stipulated above, we have

$$\mu = \frac{\mu_0 e^{\beta p}}{3(C_2/t_0)(p_1 - p) + 1} \tag{25}$$

We write Eq. (19) in differential form as follows:

$$Q\,dL = \frac{C_1}{\nu}\,dp \tag{26}$$

Using μ and ν interchangeably owing to the slight change in density within the flow path, we have

$$Q\,dL = \frac{C_1}{\nu_0}\,e^{-\beta p}\left[\frac{3C_2}{t_0}(p_1 - p) + 1\right]dp \tag{27}$$

This differential equation may be solved by the use of fundamental integrals, as follows:

$$QL = \int_{p_2}^{p_1} \frac{3C_1C_2p_1}{\nu_0 t_0}\,e^{-\beta p}\,dp - \int_{p_2}^{p_1} \frac{3C_1C_2p}{\nu_0 t_0}\,e^{-\beta p}\,dp + \int_{p_2}^{p_1} \frac{C_1}{\nu_0}\,e^{-\beta p}\,dp$$

$$\int_{p_2}^{p_1} \frac{3C_1C_2p_1}{\nu_0 t_0} = \left[-\frac{1}{\beta}\left(\frac{3C_1C_2p_1}{\nu_0 t_0}\right)e^{-\beta p}\right]_{p_2}^{p_1}$$

$$\int_{p_2}^{p_1} \frac{3C_1C_2p}{\nu_0 t_0}\,e^{-\beta p}\,dp = \left[\frac{3C_1C_2}{\nu_0 t_0}\left(\frac{p}{\beta}e^{-\beta p} + \frac{1}{\beta^2}\,e^{-\beta p}\right)\right]_{p_2}^{p_1}$$

$$\int_{p_2}^{p_1} \frac{C_1}{\nu_0}\,e^{-\beta p}\,dp = \left[-\frac{1}{\beta}\frac{C_1}{\nu_0}\,e^{-\beta p}\right]_{p_2}^{p_1}$$

To simplify the result, we substitute $p_2 = 0$ (atmospheric terminal pressure) and obtain

$$Q = \frac{C_1}{L\nu_0}\frac{1}{\beta}\left[(1 - e^{-\beta p_1}) + \frac{3C_2 p_1}{t_0} - \frac{1}{\beta}\left(\frac{3C_2}{t_0}\right)(1 - e^{-\beta p})\right] \tag{28}$$

Dimensional units stipulated for Eqs. (8), (17), and (18) are used in Eq. (28). The coefficient C_1 is given in the following:
For circular pipes;

$$C_1 = 922{,}000D^4$$

For parallel plates;

$$C_1 = 3{,}132{,}000b^3w$$

For annular spaces:

$$C_1 = 9{,}833{,}000b^3D$$

Equation (28) will permit computation of flow through capillary openings under specified conditions of pressure and temperature. Caution is indicated in the use of this formula so as not to extend it beyond the range of pressures and temperatures for which it was developed. It should be remembered that it is based on an average temperature coefficient of viscosity, valid for a limited temperature range only and independent of pressure, and is also based on the stipulation that the expression $(C_2/t_0)(p_1 - p)$ is so small, compared with unity, that all its powers greater than the first may be neglected. It has also been stipulated that the pressure coefficient β and the density ρ are constant and independent of temperature and pressure, which is correct only in a limited sense. Equation (28) will fairly describe conditions in capillary passages for oils as shown in the chart of Fig. 4 over temperature ranges from 70 to 130°F and pressures up to 5,000 psi.

An analysis of Eq. (28) within these limits shows that the effects of pressure and temperature on viscosity largely compensate for each other; as a matter of fact, when $3C_2 = \beta t_0$, Eq. (28) will revert to Eq. (19), which means that the initial increase in viscosity at any pressure within the valid range will be compensated for entirely by the decrease due to temperature rise in the flow path.

Results obtained by the use of this analysis will be modified by heat transfer from the oil to the parts of the mechanism. This will result in some reduction of temperature rise and flow capacity.

7. Viscous Flow through Eccentric Annular Spaces. In Eq. (18) we have given an expression for viscous flow through the annular space as shown in Fig. 22. This formula is based on the assumption that b is constant, or in other words, that the two circles formed by piston and cylinder are concentric. In the following, we shall develop the expression

for the assumption that the circles are not concentric; which may be the case in a good many applications.

Referring to Fig. 23, let y denote the width of the annulus at any specified angle ∂ of the radius R of the outer circle. The radius of the inner circle is r, and its center is displaced by an amount e. Then we have

$$y = R - (r \cos \gamma + e \cos \partial) \tag{29}$$

since angle γ is extremely small throughout the entire range, we may write

$$y = R - (r + e \cos \partial) \tag{30}$$

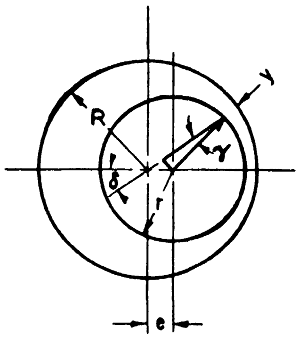

FIG. 23. Viscous flow through eccentric annular space.

From Eq. (17)

$$dQ = Cy^3R\, d\partial \tag{31}$$

where

$$C = \frac{3,131,561(p_1 - p_2)}{\nu L}$$

or

$$dQ = C[R - (r + e \cos \partial)]^3 R\, d\partial \tag{32}$$

We stipulate

$$R - r = b$$

Then

$$Q = C \int_0^{2\pi} (b - e \cos \partial)^3 R\, d\partial \tag{33}$$

If we establish the term $e/b = \epsilon =$ relative eccentricity, we have

$$Q = C \int_0^{2\pi} b^3(1 - \epsilon \cos \partial)^3 R\, d\partial \tag{34}$$

or

$$Q = CRb^3 \int_0^{2\pi} [1 - 3\epsilon \cos \partial + 3\epsilon^2 \cos^2 \partial - \epsilon^3 \cos^3 \partial]\, d\partial \tag{35}$$

Integration and substitution of limits furnishes

$$Q = 2\pi Rb^3C(1 + 1\tfrac{1}{2}\epsilon^2) \tag{36}$$

or

$$Q = \frac{9,833,102(p_1 - p_2)b^3D}{\nu L}(1 + 1\tfrac{1}{2}\epsilon^2) \tag{37}$$

If $\epsilon = 0$, both circles are concentric, and Eq. (37) reverts to Eq. (18). At maximum value of $e = b$, ϵ becomes 1; in this case the leakage is $2\frac{1}{2}$ times that of the concentric configuration.

8. Flow through Capillary Spaces with Moving Boundary. In preceding sections of this chapter we assumed that the oil at the boundaries of the capillary flow path was at rest. This is not always the case. As a

matter of fact, in reciprocating pistons, rotating shafts, and rotors there is a movement of at least one of the boundaries of the flow path. When this condition exists, equations previously developed must be modified to care for the additional transportation of fluid by the shear forces and heating by viscous shear, or the reduction in the flow through the passage, as the case may be.

We shall proceed with the flow between parallel plates and develop equations for this configuration. Flow through annular spaces may then be readily adapted by replacing the quantity w with πD.

Returning to the differential Eq. (10), we have obtained by integration

$$u = \frac{1}{\mu}\frac{dp}{dx}\frac{y^2}{2} + A_1 y + A_2 \tag{11}$$

We now stipulate that u is zero for $y = -b/2$ and $\pm S$ for $y = b/2$. S is the speed of the boundary and is positive or negative according to its direction with or against the flow of fluid.

We then have

$$\frac{1}{\mu}\frac{dp}{dx}\frac{b^2}{8} - A_1\frac{b}{2} + A_2 = 0$$

$$\frac{1}{\mu}\frac{dp}{dx}\frac{b^2}{8} + A_1\frac{b}{2} + A_2 = \pm S$$

From this we obtain

$$A_2 = \pm\frac{S}{2} - \frac{1}{\mu}\frac{dp}{dx}\frac{b^2}{8} \qquad \text{and} \qquad A_1 = \pm\frac{S}{b}$$

Then

$$u = -\frac{1}{2\mu}\frac{dp}{dx}\left(\frac{b^2}{4} - y^2\right) \pm \frac{Sy}{b} \pm \frac{S}{2} \tag{38}$$

The total flow is

$$Q = w\int_{-b/2}^{+b/2} u\,dy \tag{39}$$

Integration, substitution of limits, and simplification furnishes

$$Q = \left(-\frac{b^3}{12\mu}\frac{dp}{dx} \pm \frac{Sb}{2}\right)w \tag{40}$$

With

$$\frac{dp}{dx} = -\frac{p_1 - p_2}{L}$$

$$Q = \left[\frac{(p_1 - p_2)b^3}{12\mu L} \pm \frac{Sb}{2}\right]w \tag{41}$$

Again expressing Q in dimensional units corresponding to those in Eq. (17), we have, with S in feet per second

$$Q = \left[\frac{3{,}131{,}561(p_1 - p_2)b^3}{\nu L} \pm 360Sb\right] w \quad \text{cu in. per min} \tag{42}$$

Equation (42) indicates that there may be a great increase or decrease in flow through the leakage path, when high boundary speeds are present. Loss of power will occur due to both leakage and shear resistance. We may compute this loss as follows.

Power loss due to shear resistance is the product of shear resistance at the moving boundary and the boundary speed. We determine the value of the shear stress at the moving boundary:

$$\tau = \mu \frac{du}{dy}$$

From Eq. (38) we have

$$\tau = y \frac{dp}{dx} \pm \mu \frac{S}{b} \tag{43}$$

and for $y = b/2$,

$$\tau = \frac{b}{2} \frac{dp}{dx} + \mu \frac{S}{b} = \pm \mu \frac{S}{b} - \frac{(p_1 - p_2)}{L} \frac{b}{2} \tag{44}$$

$$\text{Shear force} = \pm \mu \frac{S}{b} wL - (p_1 - p_2) w \frac{b}{2}$$

Power loss due to shear resistance is

$$P_1 = \pm \left[\pm \mu \frac{S}{b} wL - (p_1 - p_2) w \frac{b}{2}\right] S \tag{45}$$

Added to this is the power loss due to capillary leakage:

$$P_2 = Q(p_1 - p_2) = \frac{(p_1 - p_2)^2 b^3 w}{12\mu L} \pm (p_1 - p_2) w \frac{b}{2} S \tag{46}$$

$$P_1 + P_2 = P = \mu \frac{S^2}{b} wL + \frac{(p_1 - p_2)^2 b^3 w}{12\mu L} \tag{47}$$

The power loss is the same, regardless of the direction of boundary speed.

Expressing P in the same units as Q in Eq. (42), we have

$$P = \frac{\nu S^2 wL}{72.3b} + \frac{3{,}131{,}561(p_1 - p_2)^2 b^3 w}{\nu L} \quad \text{in.-lb per min} \tag{48}$$

9. Flow with Moving Boundary and Variable Viscosity. In the preceding section, as in Sec. 4, equations for viscous flow were developed with

the assumption that the viscosity in the flow path is constant. To arrive at practically useful results, it will again be necessary to consider the changes in viscosity due to both pressure and temperature changes.

In Sec. 6 a simple relationship was found to exist between the pressure drop and temperature increase in the flow path, which made it possible to set up a differential equation and integrate it to establish the flow as function of pressure with variable viscosity.

With energy imparted to the fluid by both pressure flow and viscous shear, the temperature rise cannot be expressed as function of the pressure drop only, and a complex relationship exists, which defies attempts at analytical solution. To arrive at an approximate solution, the suggestion has been made to compute Q by using an average value of the viscosity between entrance and exit of the flow path,[1] μ_{avg} being $(\mu_1 + \mu_2)/2$. A simple analysis will show that this method will not produce even approximately correct results, except in cases of very slight variations in viscosity.

We have seen that in the pressure and temperature range under consideration μ or ν may be expressed as follows:

$$\nu = \nu_0 e^{\beta p} \left(\frac{t_0}{t}\right)^K \tag{49}$$

This indicates a nonlinear drop of ν in the flow path, even if pressure drop and temperature rise were linear. As a matter of fact, referring to Eq. (42), and remembering the continuity law, one concludes that the pressure gradient cannot be constant, but must be proportional to ν. The temperature increase may be assumed to be proportional to the power absorbed, as expressed in Eq. (47), which, in turn, is proportional to viscosity and pressure drop. These factors contribute to a rapid initial drop in viscosity in the flow path.

Any average value for the viscosity in the flow path should therefore be weighed heavily in favor of the lower values. To this end, it is suggested that the average viscosity be defined as that existing at a mean value of pressure and temperature in the flow path, or

$$\nu_{avg} = \nu_0 e^{\beta p_1/2} \left[\frac{t_0}{t_0 + (\Delta t/2)}\right]^3 \tag{50}$$

p_2 or terminal pressure is again assumed to be zero. The temperature rise Δt may be computed from Eqs. (42) and (48) as follows:

$$\Delta t = \frac{P}{Q} C_2 \qquad \text{or} \qquad \Delta t = \frac{P}{Q} \times 7.2 \times 10^{-3} \tag{51}$$

[1] See bibliography at end of chapter.

Rather than to attempt computation of ν_{avg} by algebraic manipulation, it will be found easier to do this by a process of trial and error. Substitution of likely values of ν_{avg} in Eqs. (42) and (48) will permit computation of Δt, which may then be used to obtain more accurate values of ν_{avg}. Generally, after a few trials, corrected values, sufficiently accurate for the purpose intended, may be obtained. The following example will illustrate this procedure.

Example 1: We assume that the $\frac{3}{4}$-in. piston discussed at the end of Sec. 5 is traveling at a speed of 3 ft per sec. Operating temperature t_0 is 120°F. If the velocity is in direction of the pressure flow, both pressure flow and shear transport add, and the temperature increase is relatively small, and its influence on the viscosity will be partly offset by the initial increase due to the pressure coefficient. By making $\nu_{avg} = \nu_0 = 40$ centistokes, we obtain as first approximation from Eqs. (42) and (48)

$$Q = 0.95 + 1.2 = 2.15 \text{ cu in. per min}$$
$$P = 1{,}015 + 1{,}900 = 2{,}915 \text{ in.-lb per min}$$
$$\Delta t = \frac{P}{Q} \times 7.2 \times 10^{-3} = 9.75°\text{F}$$

$$\nu_{avg} = 44 \qquad \text{with } \beta_{approx} = 2 \times 10^{-4}$$
$$Q_{corr} = 2.06 \text{ cu in. per min}$$
$$P_{corr} = 2{,}860 \text{ in.- lb per min}$$
$$\Delta t_{corr} = 10°\text{F}$$

If the velocity is against the direction of pressure flow, the resultant flow is the difference between pressure and shear transport flow and will assume a very small value, in consequence of which there is a great increase in temperature, causing corresponding variation in viscosity in the flow path.

By trying a few values of ν_{avg}, it will be found that Eqs. (42) (48), and (50) are satisfied by the following values:

$$Q = 0.41 \text{ cu in. per min}$$
$$P = 3{,}810 \text{ in.-lb per min}$$
$$\nu_{avg} = 23.5 \text{ centistokes}$$
$$\Delta t = 67°\text{F}$$

Determination of leakage and power losses with the aid of the analysis developed in this chapter may be used to advantage in the design of hydraulic machinery to obtain operation at maximum efficiency. The following example will illustrate this.

Example 2: Assuming that the $\frac{3}{4}$-in. piston discussed above serves as operating plunger in a rotary pump, what viscosity of oil would result in minimum power loss? The plunger reciprocates, discharging oil from the cylinder during the pressure stroke and taking in oil during the suction stroke. During the former, pressure leakage takes place, which is opposed by shear transport of oil due to the piston traveling against the direction of pressure. On the suction stroke, there is shear transport of oil from the cylinder toward the outside. Power loss occurs owing both to leakage and viscous drag on the pressure stroke and to viscous drag on suction. Power loss

on the pressure stroke predominates, and we shall establish that average viscosity at which this loss becomes a minimum.

We form the derivative of P in Eq. (48):

$$\frac{dP}{d\nu} = \frac{S^2wL}{72.3b} - \frac{3{,}131{,}561(p_1 - p_2)^2b^3w}{\nu^2L} = 0$$

whence

$$\nu = \frac{(p_1 - p_2)b^2}{SL} \times 15{,}000 \tag{52}$$

Substituting ν from Eq. (52) in Eq. (48), we obtain

$$P = 416(p_1 - p_2)Swb \tag{53}$$

Similarly

$$Q = -152Swb \tag{54}$$

With

$$\begin{aligned} p_1 - p_2 &= 2{,}000 \\ b &= 4.75 \times 10^{-4} \\ S &= 3 \text{ ft per sec} \\ L &= 1/24 \text{ ft} \end{aligned}$$

we have

$$\nu_{\text{avg}} = 54 \text{ centistokes}$$

We may compute Δt from Eq. (51), but must remember that the negative sign of Q indicates flow against the pressure gradient, and part of the power loss is recovered and, therefore, not converted into heat. We have

$$\Delta t = \frac{P - |Q|(p_1 - p_2)}{|Q|} \times 7.2 \times 10^{-3} \tag{55}$$

By substituting expressions for P and Q from Eqs. (53) and (54), we obtain

$$\Delta t = \frac{264}{152}(p_1 - p_2) \times 7.2 \times 10^{-3} \tag{56}$$

$$\Delta t = 25° \text{ F}$$
$$\nu = 60 \text{ centistokes at } 120° \text{ F}$$

Referring to Fig. 4, this indicates an oil of 500 SSU at 100°F.

$$P = 2{,}800 \text{ in.-lb per min}$$
$$Q = -0.515 \text{ cu in. per min}$$

Added to this is the power loss on the suction stroke. This equals

$$P = \frac{\nu S^2wL}{72.3b} \tag{57}$$

Here ν is the average viscosity on the suction stroke. Also

$$Q = 360Sbw \tag{58}$$

Average viscosity on suction stroke is approximately 55 centistokes. Hence

$$\begin{aligned} P &= 1{,}410 \text{ in.-lb per min} \\ Q &= 1.2 \text{ cu in. per min} \\ \Delta t &= 8.5°\text{F} \\ \text{Total power loss} &= 4{,}210 \text{ in.-lb per min} \end{aligned}$$

At a pressure $p_1 - p_2 = 1{,}000$ psi, ν_{avg} becomes 27 centistokes, and

$$\begin{aligned} \Delta t &= 12\tfrac{1}{2}°\text{F} \\ \nu_0 &= 28 \text{ centistokes (about 250 SSU at 100°F)} \end{aligned}$$

These figures agree well with the manufacturers' recommendations for oils to be used in this type of pump.

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Acknowledgement is made of the work of the following authors upon which much of the analysis in this chapter was based.

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CHAPTER VI

VISCOUS AND TURBULENT FLOW

1. Turbulent Flow; Darcy's Formula. We have shown in the preceding chapter that there are two distinct and entirely different types of flow: viscous or laminar flow and turbulent flow. The laws governing viscous flow were derived, and the expressions for velocities and pressure drop for this type of flow were developed.

We have shown in Eq. (6), Chap. V, that the average velocity V in a circular pipe is

$$V = \frac{p_1 - p_2}{32\mu L} D^2$$

From this we may compute

$$p_1 - p_2 = \frac{32\mu L V}{D^2} \tag{1}$$

If the friction head in the pipe is

$$h_f = \frac{p_1 - p_2}{w}$$

and

$$w = \rho g$$

we have

$$h_f = \frac{32\mu L V}{\rho g D^2} \tag{2}$$

The friction head in a pipe for laminar flow, therefore, is proportional to the velocity of flow.

For the computation of friction loss in turbulent flow, empirical formulas must be resorted to, the most generally accepted of which is Darcy's formula:

$$h_f = f\frac{L}{D}\frac{V^2}{2g} \tag{3}$$

It should be noted that in turbulent flow, the friction head is proportional to the square of the velocity of flow.

The coefficient f is empirical, and many experimenters have collected data from which this coefficient may be computed. Before dealing with these data, consideration will be given to the development of the criterion that determines whether flow is laminar or turbulent.

2. The Reynolds Number. To develop this criterion, we will bring Eqs. (2) and (3), which describe the two forms of flow, into a form common to both. Darcy's formula is written as follows:

$$h_f = f \frac{L}{D} \frac{V^2}{2g}$$

We may obtain a similar expression of the Hagen-Poiseuille formula by multiplying it by V/V. Then we have

$$h_f = \frac{32\mu L V^2}{V \rho g D^2} \tag{4}$$

or

$$\frac{64\mu}{\rho V D} \frac{L}{D} \frac{V^2}{2g} \tag{5}$$

Equation (5) shows that the coefficient f in Darcy's formula has its counterpart in the expression $64\mu/\rho VD$ in the equation for viscous flow.

The expression $\rho VD/\mu$ is a dimensionless figure, as may readily be shown by substituting dimensions for the quantities involved, with ρ in slugs per cu ft, V in ft per sec, D in ft, μ in lb-sec per sq ft. This nondimensional combination of quantities is known as the "Reynolds number," named after Osborne Reynolds. It will be designated by the symbol R, and we have

$$R = \frac{\rho VD}{\mu} = \frac{VD}{\nu} \tag{6}$$

The Reynolds number is the criterion whether flow is viscous or turbulent. At the so-called "critical value" of the Reynolds number, flow changes from laminar to turbulent. The early experiments of Osborne Reynolds and later investigations by others proved the existence of this critical value.

It has been found that there is no sudden transition from laminar to turbulent flow, but a gradual change beginning with slight initial disturbances and passing through a transitional range until complete turbulence exists. The value of the critical Reynolds number below which laminar flow is certain to exist, has been found as 2,000. Laminar flow has been found to exist at much higher values of R, but the critical value of the Reynolds number definitely establishes the point below which turbulent flow may not exist.

The Hagen-Poiseuille law applies to flow of fluids for Reynolds numbers below 2,000. Above the critical Reynolds number, Darcy's formula applies. The friction factor f in Darcy's formula has been shown to

be a function of the Reynolds number R and the roughness of the surface of the pipe or conduit. This is true only for turbulent flow, as in laminar

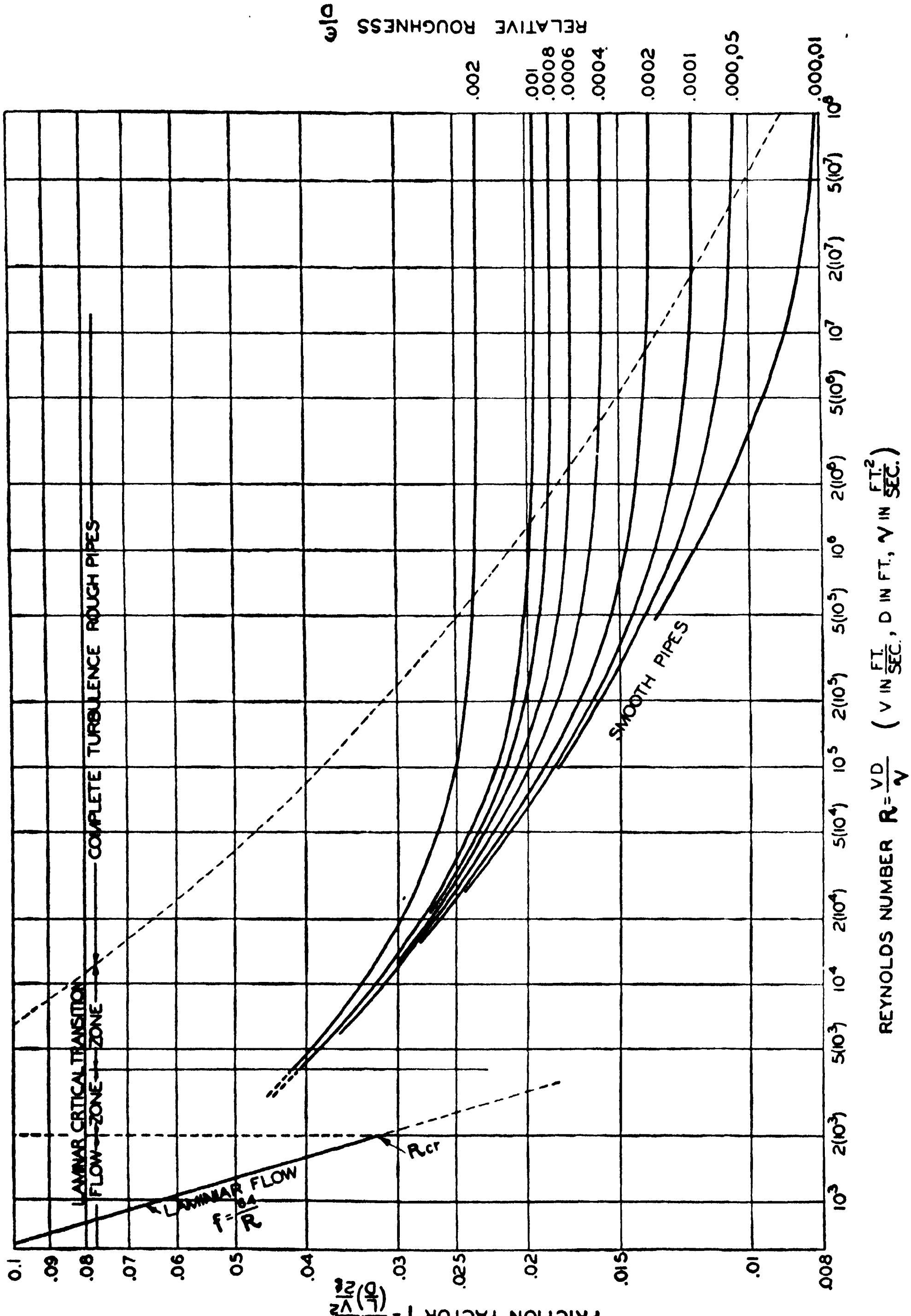


FIG. 24. Stanton diagram.

flow, f is independent of the roughness of the surface to a large degree and is equal to the coefficient: $64/R$ in the Hagen-Poiseuille equation.

3. The Stanton Diagram. The value of f may best be represented in a graph and as a function of R. A diagram showing this relationship is

known as the "Stanton diagram," because Stanton was the first to employ this representation of the friction factor.

A chart taking advantage of the functional relationships established in recent years has been drawn up by Lewis F. Moody[1] and is reproduced here, with the author's permission, in a form convenient for the reader of this text. In Fig. 24 friction factor f is shown as function of R and the relative roughness ϵ/D (ϵ being a linear quantity in feet representing the

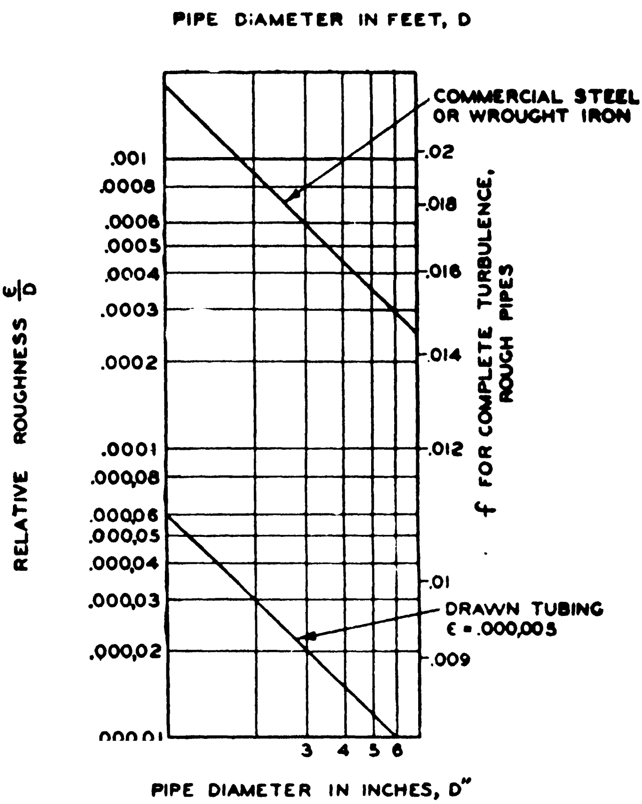


FIG. 25. Relative roughness as function of pipe diameter.

absolute roughness). An auxiliary chart is given in Fig. 25 from which ϵ/D can be taken for any size and type of pipe.

In Fig. 24 all quantities given are in the fundamental units of the English gravitational system. For convenience in using the charts, the Reynolds number will be expressed in more commonly used units as follows:

If V = velocity, ft per sec
D = pipe diameter, in.
ν = kinematic viscosity, centistokes

then

$$R = \frac{92{,}900}{12}\frac{VD}{\nu} = 7{,}740\frac{VD}{\nu} \tag{7}$$

[1] Lewis F. Moody, Friction Factors for Pipe Flow, *Trans. ASME*, **66**, p. 671, 1944.

With friction factor f thus found from Fig. 24, the loss of head, h_f, may be computed from Darcy's formula as follows:

$$h_f = f \frac{L}{D} \frac{V^2}{2g}$$

With $p_f = h_f w$, we have

$$p_f = f \frac{L}{D} \frac{V^2}{2} \frac{w}{g} \tag{8}$$

$$= f \frac{L}{D} \frac{V^2}{2} \frac{62.4s}{g} \tag{9}$$

$$= 0.9698 f \frac{L}{D} V^2 s \tag{10}$$

Expressed in convenient units, we have, with

p_f in psi
L in ft
D in in.
V in ft per sec

$$p_f = \frac{0.9698 \times 12}{144} f \frac{L}{D} V^2 s = 0.0808 f \frac{L}{D} V^2 s \tag{11}$$

The use of the formulas and charts for the determination of friction loss in pipes will be illustrated in the following explanation and example.

a. Determine the viscosity of the fluid in centistokes. For hydraulic oils of various viscosities this may be done by referring to Fig. 4.

b. Determine the Reynolds number R from Eq. (7).

c. Determine relative roughness from Fig. 25.

d. Determine friction factor f from Fig. 24 by following the line corresponding to ϵ/D to the value of the Reynolds number R.

e. Compute p_f from Eq. (11).

Example 1: Determine friction loss for Vacuum oil DTEBB flowing through a 1-in.-ID pipe, 50 ft long, at a velocity of 20 ft per sec at 120°F.

a. Viscosity of DTEBB oil at 120°F = 110 centistokes

b. $R = 7{,}740 \dfrac{VD}{\nu} = \dfrac{7{,}740 \times 20 \times 1}{110} = 1{,}410$

c. Since Reynolds number is below 2,000, roughness of the pipe does not enter into the calculation.

d. $f = 0.045 = {}^{64}\!/\!_{1410}$

e. $p_f = 0.0808 \times 0.045 \times 50 \times 400 \times 0.91 = 60$ psi.

Example 2: Determine friction loss of Vacuum DTE oil "Light" at 100°F, flowing through a 2-in.-ID commercial steel pipe, 100 ft long, at a speed of 30 ft per sec.

a. Viscosity of Vacuum DTE light oil at 100°F = 32 centistokes

b. Reynolds number $R = 7{,}740 \dfrac{VD}{\nu} = \dfrac{7{,}740 \times 30 \times 2}{32} = 14{,}500$

c. Relative roughness ϵ/D from Fig. 25 = 0.0009

d. Friction factor f from Fig. 24 = 0.03

e. $p_f = \dfrac{0.0808 \times 0.03 \times 100 \times 900 \times 0.91}{2} = 100$ psi

4. Other Losses in Pressure. In addition to the losses of pressure due to velocity and friction head in straight pipe, other losses occur in a hydraulic circuit, which will be dealt with in the following.

a. Sudden Enlargement. If the cross section of a pipe is suddenly enlarged, as shown in Fig. 26, the velocity will be suddenly reduced from V_1 to V_2, and loss of pressure will result from the turbulence caused by the impact of the relatively fast moving stream in the small pipe upon the relatively slow moving stream in the large pipe.

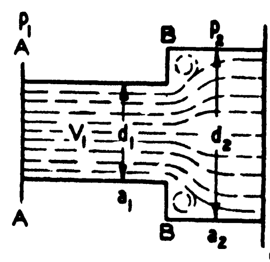


FIG. 26. Sudden enlargement.

Bernoulli's equation between points 1 and 2 furnishes

$$\frac{p_1}{w} + \frac{V_1^2}{2g} = \frac{p_2}{w} + \frac{V_2^2}{2g} + \text{loss} \tag{12}$$

whence

$$\text{Loss} = \frac{V_1^2}{2g} - \frac{V_2^2}{2g} - \left(\frac{p_2}{w} - \frac{p_1}{w}\right) \tag{13}$$

If the areas of the small and large pipes are denoted as A_1 and A_2, respectively, we have

$$A_1V_1 = A_2V_2$$

Velocity V_1 may be presumed to exist at cross section BB, so that pressure p_1 will exist at that cross section. Therefore the total pressure at section BB will be A_2p_1, and at section CC, A_2p_2. p_2 must be larger than p_1 owing to the loss in velocity head between sections BB and CC.

Therefore a force $A_2p_2 - A_2p_1$ exists, which causes deceleration of the fluid between sections BB and CC, or

$$A_2p_2 - A_2p_1 = \frac{V_1 - V_2}{t} M \tag{14}$$

where M is the mass of fluid, in slugs, between points BB and CC. Since

$$\frac{M}{t} = A_2V_2\rho = A_2V_2\left(\frac{w}{g}\right) \tag{15}$$

we have

$$\frac{p_2}{w} - \frac{p_1}{w} = \frac{V_2}{g}(V_1 - V_2) \tag{16}$$

Combining (16) and (13), we have

$$\text{Loss} = \frac{V_1^2}{2g} - \frac{V_2^2}{2g} - \frac{V_2}{g}(V_1 - V_2) = \frac{(V_1 - V_2)^2}{2g} \tag{17}$$

If we substitute for V_1 its equivalent $(A_2/A_1)V_2$, we have

$$h_f = \left(\frac{A_2}{A_1} - 1\right)^2 \frac{V_2^2}{2g} \tag{18}$$

If discharge takes place into a reservoir of large area, V_2 becomes zero and the loss is

$$h_f = \frac{V_1^2}{2g} \tag{19}$$

This means that when discharge takes place into a large reservoir or pressure vessel, the entire velocity head is lost in turbulence.

b. Sudden Contraction. In the case of a sudden contraction, a simple mathematical analysis is not possible. Experimental investigation has shown that the loss may be expressed in the form

$$h_f = K\frac{V^2}{2g} \tag{20}$$

V being the velocity in the smaller pipe. The coefficient K depends on the ratio of the pipe diameters. Table I has been compiled by George E. Russell.[1]

TABLE I

$\frac{d_1}{d_2}$	K
4	0.45
3.5	0.43
3	0.42
2.5	0.40
2.0	0.37
1.5	0.28
1.25	0.19
1.1	0.10
1.0	0

A special case of sudden contraction is the entrance loss from a reservoir into a pipe line. In this case the coefficient K becomes 0.5.

[1] George E. Russell, "Hydraulics," 5th ed., p. 203, Henry Holt and Company, Inc., New York, 1942.

c. Bends and Elbows. Losses in bends and elbows may again be expressed in the form

$$h_f = K \frac{V^2}{2g}$$

For pipe bends the coefficient K is approximately 0.3 for radii between 2½ and 5 pipe diameters.

For screwed elbows, up to 2 in., K equals about 0.72. For right-angle bends, $K = 1.20$; for 45° bends, $K = 0.263$. For T's, K equals about 1.50.[1]

d. Valves. The coefficients given in Table II apply to valves of customary design.[2]

TABLE II

Valve	Dimension, in.	Coefficient K
Globe...........	1*	5.3 (flow against bottom of seat) 4.4 (flow against top of seat)
Globe...........	1½†	16.5 (flow against bottom of seat) 12.5 (flow against top of seat)
Angle...........	1¼	2.1 (flow against bottom) 2.6 (flow against top)
Gate...........	1½	0.3

* Diameter.
† Short end-to-end dimension.

5. Expression of Losses in Equivalent Length of Pipe. Darcy's formula for flow in straight length of pipe is very similar in construction to the mathematical expression for the losses in bends, valves, and fittings.

Rather than computing the loss in feet for each individual item in a hydraulic circuit, it may be preferable to express individual losses in fittings, etc., in equivalent length of straight piping, having the same loss as the particular fitting. To do this, we equate Darcy's formula and the fitting-loss formula:

$$f \frac{L}{D} \frac{V^2}{2g} = \frac{KV^2}{2g} \tag{21}$$

[1] A. H. Gibson, "Hydraulics and Its Application," 4th ed., pp. 251–257, D. Van Nostrand Company, Inc., New York, 1930.

[2] Hütte, "Des Ingenieurs Taschenbuch," 23d ed., p. 306. W. Ernst und Sohn, Berlin, 1920.

From this we obtain the equivalent length

$$L_e = \frac{K}{f} D \qquad \text{if } D \text{ in feet} \tag{22}$$

$$L_e = \frac{K}{f} \frac{D}{12} \qquad \text{if } D \text{ in inches} \tag{23}$$

From this formula we may see that the equivalent length may be computed for a given size of fitting by first determining the friction coefficient f in Darcy's formula for the corresponding straight pipe.

The literature contains numerous references to equivalent length of pipe based on arbitrary values of f. These figures should be used with great caution, as the coefficient f is subject to great variations, especially in laminar flow, which is common in oil-pressure work.

6. Divided Flow. When flow of fluid is diverted from a common main to a number of branch lines, the resistance may be computed in the following manner. Darcy's formula [Eq. (11)] may be written

$$p_f = \phi V^2 \tag{24}$$

in which ϕ denotes a composite coefficient equaling 0.0808 $(L/D)fs$. If V is the velocity in an imaginary pipe that produces the same effective pressure loss as the actual branch conduits, we have, in the case of two branches,

$$p_f = \phi_1 V_1^2 = \phi_2 V_2^2 = \phi V^2 \tag{25}$$

The continuity equation furnishes

$$A_1 \sqrt{\frac{p_f}{\phi_1}} + A_2 \sqrt{\frac{p_f}{\phi_2}} = A \sqrt{\frac{p_f}{\phi}} \tag{26}$$

or

$$\frac{A_1}{\sqrt{\phi_1}} + \frac{A_2}{\sqrt{\phi_2}} = \frac{A}{\sqrt{\phi}} \tag{27}$$

From this we obtain

$$\phi = \left(\frac{A\sqrt{\phi_1\phi_2}}{A_1\sqrt{\phi_2} + A_2\sqrt{\phi_1}}\right)^2 \tag{28}$$

The effective pressure drop, therefore, will be

$$p_f = \left(\frac{AV\sqrt{\phi_1\phi_2}}{A_1\sqrt{\phi_2} + A_2\sqrt{\phi_1}}\right)^2 \tag{29}$$

In order to compute p_f, the coefficients ϕ_1 and ϕ_2 must first be determined. To this end an approximate computation of the Reynolds num-

ber must first be made to determine whether the flow is viscous or turbulent. This may be done by estimating approximately the velocities V_1 and V_2. If the flow is viscous, the velocities V_1 and V_2 may be more accurately determined as follows:

From Eq. (2) we have

$$\frac{32\mu L_1 V_1}{\rho g D_1^2} = \frac{32\mu L_2 V_2}{\rho g D_2^2} \tag{30}$$

or

$$\frac{L_1 V_1}{A_1} = \frac{L_2 V_2}{A_2} \tag{31}$$

From the continuity equation we obtain

$$V_1 = \frac{AV - A_2 V_2}{A_1} \tag{32}$$

Hence

$$\frac{L_1(AV - A_2 V_2)}{A_1^2} = \frac{L_2 V_2}{A_2} \tag{33}$$

from which

$$V_2 = \frac{L_1 A V}{(A_1^2 L_2/A_2) + L_1 A_2} \tag{34}$$

and similarly

$$V_1 = \frac{L_2 A V}{(A_2^2 L_1/A_1) + L_2 A_1} \tag{35}$$

With V_1 and V_2 determined, we may compute the Reynolds numbers R_1 and R_2 and friction coefficients f_1 and f_2. Then

$$\phi_1 = 0.0808 \frac{L_1}{D_1} f_1 s \tag{36}$$

and

$$\phi_2 = 0.0808 \frac{L_2}{D_2} f_2 s \tag{37}$$

In case of turbulent flow, coefficients f_1 and f_2 may be taken from Fig. 24. Other losses may be cared for by substituting equivalent lengths L_{e_1} and L_{e_2} in the expressions for ϕ_1 and ϕ_2.

7. Computation of Losses in a Hydraulic Circuit. To facilitate understanding of the losses and substitution of equivalent length, an example will be given in the following, covering a hydraulic circuit as shown in Fig. 27.

Example: The circuit is fed by a hydraulic pump having a capacity of 10 gpm. Oil is Vacuum DTEBB with a viscosity of 110 centistokes at 120°F. The inside diameters of the pipes and their lengths are indicated in Fig. 27. To be strict, the length of fittings or values in the straight lengths of pipe should be deducted, but the calculation at best can be only approximate, and extreme accuracy in these computations is not warranted.

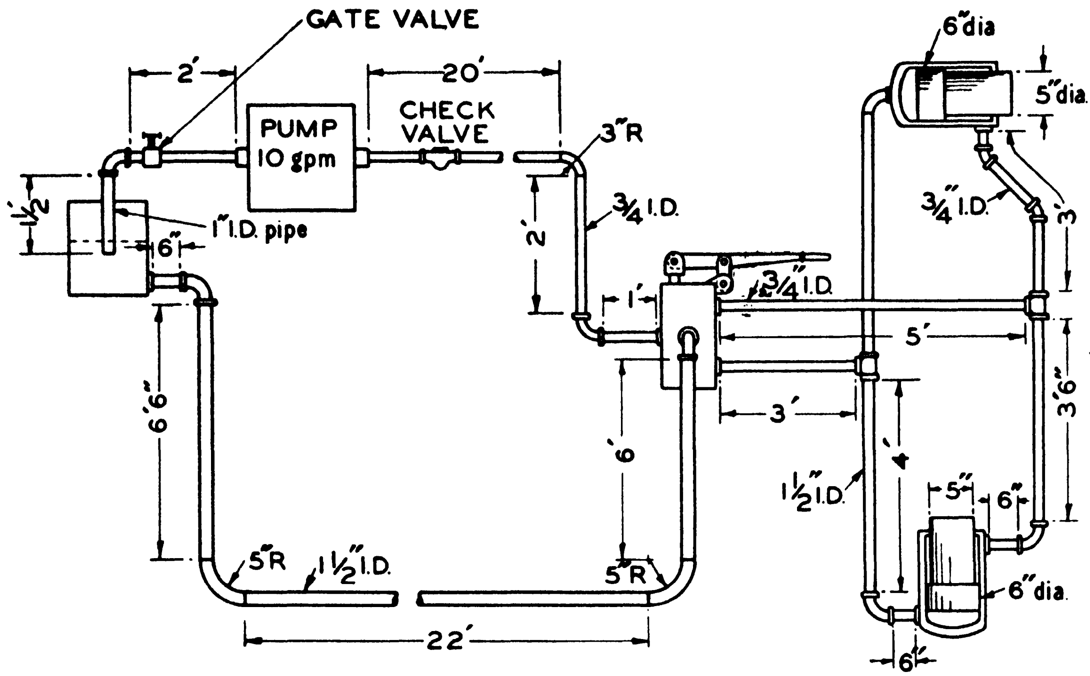


Fig. 27. Hydraulic circuit.

Analysis of Losses. a. Suction Line.

Inside diameter = 1 in.
Pump capacity = 10 gpm

$$V = \frac{10 \times 231}{12 \times 60 \times 0.785} = 4.1 \text{ ft per sec}$$

$$R = 7{,}740 \frac{VD}{\nu} = \frac{7{,}740 \times 4.1 \times 1}{110} = 290$$

$$f = {}^{64}\!/_{290} = 0.22$$

Losses:

Straight length of pipe			= 3.50 ft
Entrance loss:	$K = 0.5$	$L_e = \dfrac{0.5}{0.22}\,\dfrac{1}{12}$	= 0.19 ft
Elbow:	$K = 0.72$	L_e	= 0.27 ft
Gate valve:	$K = 0.3$	L_e	= 0.115 ft
			4.075 ft

Total loss in suction line:

$$p_f = \frac{0.0808 \times 0.22 \times 4.075 \times 16.8 \times 0.91}{1} = 1.1 \text{ psi}$$

b. Discharge Line. The hydraulic operating valve is presumed to be in its return position with pump discharge directed into the pullback line of the hydraulic cylinder.

Diameter of line = ¾ in.

$$V = \frac{10 \times 231}{12 \times 60 \times 0.44} = 7.3 \text{ ft per sec}$$

$$R = \frac{7{,}740 \times 7.3 \times 0.75}{110} = 385$$

$$f = {}^{64}\!/_{385} = 0.166$$

Losses:

Straight length of pipe up to T			= 28 ft
Check valve:	$K = 16.5$	$L_e = \frac{16.5}{0.166}\frac{0.75}{12}$	= 6.25 ft
Bend:	$K = 0.3$	L_e	= 0.11 ft
Square elbow:	$K = 1.2$	L_e	= 0.45 ft
T:	$K = 1.5$	L_e	= 0.56 ft
			35.37 ft

Total loss in discharge line exclusive of operating valve:

$$p_f = \frac{0.0808 \times 0.166 \times 35.37 \times 53.5 \times 0.91}{0.75} = 30.0 \text{ psi}$$

c. ¾-in. Branch Lines. Conditions obviously indicate the existence of viscous flow in these lines. All areas are the same, that is, $A = A_1 = A_2$. Hence

$$V_1 = \frac{L_2 V}{L_1 + L_2} = \frac{4 \times 7.3}{7} = 4.17 \qquad R_1 = 220 \qquad f_1 = 0.29$$

and

$$V_2 = \frac{L_1 V}{L_1 + L_2} = \frac{3 \times 7.3}{7} = 3.14 \qquad R_2 = 165 \qquad f_2 = 0.388$$

L_1 = straight length			= 3 ft
Two 45° L's:	$K = 0.263$	$L_e = \frac{2 \times 0.263}{0.29}\frac{0.75}{12}$	= 0.113 ft
Exit loss:	$K = 1$	L_e	= 0.216 ft
		L_{e_1}	= 3.329 ft
L_2 = straight length			= 4 ft
Square elbow:	$K = 1.20$	$L_e = \frac{1.20 \times 0.75}{0.388 \times 12}$	= 0.193 ft
Exit loss:	$K = 1$	L_e	= 0.160 ft
		L_{e_2}	= 4.353 ft

Corrected velocities and friction coefficients:

$$V_1 = \frac{4.353 \times 7.3}{7.682} = 4.14 \qquad R_1 = 218 \qquad f_1 = 0.294$$

$$V_2 = \frac{3.329 \times 7.3}{7.682} = 3.15 \qquad R_2 = 167 \qquad f_2 = 0.382$$

Then

$$\phi_1 = 0.0808 \frac{3.329}{0.75} (0.294 \times 0.91) = 0.096$$

$$\phi_2 = 0.0808 \frac{4.353}{0.75} (0.382 \times 0.91) = 0.162$$

$$\phi = \left(\frac{\sqrt{\phi_1\phi_2}}{\sqrt{\phi_1} + \sqrt{\phi_2}}\right)^2 = \left(\frac{\sqrt{0.096 \times 0.162}}{\sqrt{0.096} + \sqrt{0.162}}\right)^2 = 0.03$$

$$p_f = 7.3^2 \times 0.03 = 1.6 \text{ psi}$$

d. 1½-in. Branch Lines. Velocity in the common 1½-in. line is

$$V = \frac{10 \times 231 \times 28.3}{12 \times 60 \times 1.76 \times 8.7} = 6 \text{ ft per sec}$$

Viscous flow is again indicated, and we have

$$V_1 = \frac{L_2V}{L_1 + L_2} = \frac{4.5}{8.5} \times 6 = 3.18 \qquad R_1 = 337 \qquad f_1 = 0.19$$

$$V_2 = \frac{L_1V}{L_1 + L_2} = \frac{4}{8.5} \times 6 = 2.82 \qquad R_2 = 297 \qquad f_2 = 0.215$$

L_1 = straight length			= 4 ft
One bend:	$K = 0.3$	$L_e = \frac{0.3}{0.19} \times \frac{1.5}{12}$	= 0.197 ft
Entrance loss:	$K = 0.5$	L_e	= 0.330 ft
		L_{e_1}	= 4.527 ft
L_2 = straight length			= 4.5 ft
Square elbow:	$K = 1.2$	$L_e = \frac{1.2}{0.215} \frac{1.5}{12}$	= 0.695 ft
Entrance loss:	$K = 0.5$	L_e	= 0.290 ft
		L_{e2}	= 5.485 ft

Corrected velocities and friction coefficients:

$$V_1 = \frac{5.485 \times 6}{10.012} = 3.3 \qquad R_1 = 350 \qquad f_1 = 0.182$$

$$V_2 = \frac{4.527 \times 6}{10.012} = 2.7 \qquad R_2 = 285 \qquad f_2 = 0.225$$

Then

$$\phi_1 = 0.0808 \times \frac{4.527}{1.5} \times 0.182 \times 0.91 = 0.04$$

$$\phi_2 = 0.0808 \times \frac{5.485}{1.5} \times 0.225 \times 0.91 = 0.06$$

$$\phi = \left(\frac{\sqrt{0.04 \times 0.06}}{\sqrt{0.04} + \sqrt{0.06}}\right)^2 = 0.0120$$

$$p_f = 6^2 \times 0.0120 = 0.43 \text{ psi}$$

e. Return Line

$$V = 6 \text{ ft per sec}$$
$$R = 640$$
$$f = 0.10$$

Straight length			= 38 ft
One T:	$K = 1.5$	$L_e = \frac{1.5}{0.1} \frac{1.5}{12}$	= 1.87 ft
Two bends:	$K = 0.3$	L_e	= 0.75 ft

$$\text{Two square L's: } K = 1.2 \qquad L_e \qquad = 3.00 \text{ ft}$$

$$\text{Exit loss:} \qquad K = 1 \qquad L_e \qquad = \underline{1.25 \text{ ft}}$$

$$L_e = 44.87 \text{ ft}$$

$$p_f = \frac{0.0808 \times 0.10 \times 44.87 \times 36 \times 0.91}{1.5} = 7.9 \text{ psi}$$

F. Operating Valve. The operating valve is of such size as to accommodate the 1½-in. return line. Standard valves are made with all connections the same size, so that it is necessary to bush for the ¾-in. pipes at the valve. This will result in a case of sudden enlargement and sudden contraction. Area of the valve is generally made somewhat smaller than that of the corresponding pipe. In this case area is assumed to be 1 sq in.

The resistance coefficient may be assumed to be about 16.5 for each flow through the valve. The resistance through the valve then figures as follows:

¾-in. connection:

Velocity at intake, ¾-in. pipe = 7.3 ft per sec

$$\text{Velocity through valve} = \frac{10 \times 231}{1 \times 12 \times 60} = 3.2 \text{ ft per sec}$$

Velocity at outlet = 7.3 ft per sec

$$\text{Sudden enlargement } p_f = \frac{h_f s}{2.32} = \frac{0.91}{2.32} h_f \qquad = 0.392\, h_f$$

$$= 0.392 \frac{(V_1 - V_2)^2}{2g} = 0.103 \text{ psi}$$

$$\text{Turbulence loss} = \frac{0.392 \times 16.5 \times 10.2}{2g} \qquad = 1.03 \text{ psi}$$

$$\text{Sudden contraction } p_f = \frac{0.392 \times 0.28 \times 50}{2g} = 0.086 \text{ psi}$$

$$\text{Total loss } p_f = 1.219 \text{ psi}$$

1½-in. connection:

Velocity intake, 1½-in. pipe = V = 6.0 ft per sec
Velocity through valve = V = 10.6 ft per sec

$$\text{Sudden contraction } p_f = \frac{0.392 \times 0.20 \times 110}{2g} = 0.134 \text{ psi}$$

$$\text{Turbulence loss} = \frac{0.392 \times 16.5 \times 110}{2g} \qquad = 11.00 \text{ psi}$$

$$\text{Sudden enlargement } p_f = \frac{0.392(10.6 - 6.0)^2}{2g} = 0.13 \text{ psi}$$

$$\text{Total loss } p_f = 11.264 \text{ psi}$$

Total loss of pressure in the system, which must be supplied by the hydraulic pump:

Suction line	= 1.10
Discharge	= 30.00
Branch line	= 1.60
Return branch = $0.43 \times \frac{28.3}{8.7}$	= 1.40
Return line = $7.9 \times \frac{28.3}{8.7}$	= 25.60
Operating valve	= 1.22
Operating valve return = $11.264 \times \frac{28.3}{8.7}$	= 36.50
Total loss of pressure to be supplied by the pump	= 97.42 psi

The total loss does not include the loss in the pump itself, which is part of the pump's efficiency.

No mention has been made in the analysis of the velocity head of the fluid. The velocity head is not a loss but energy imparted to the fluid and should be recoverable. actually this energy is lost in turbulence when the fluid enters the hydraulic cylinders, as previously explained.

8. Recommended Velocities. Velocities actually used in hydraulic practice have been established by experience. Too high a velocity results in excessive pressure drop in the system, while too low a velocity increases the expense of piping and valves. The velocities given in Table III have been found satisfactory for most practical applications.

TABLE III

Part of System	*Velocity, ft per sec*
Suction lines, ½ to 1 in.	2–4
1¼ in. and up	5
Discharge lines, ½ to 2 in.	10
Over 2 in.	12
Discharge through control valves and other short restrictions	20
Relief and safety valves	100

9. Flow through Orifices. In Sec. 4, Chap. IV, we have shown that according to Toricelli's theorem the velocity of discharge from an opening under a pressure head h equals the velocity of a body falling a distance h in vacuo.

$$V = \sqrt{2gh} \tag{38}$$

The total quantity flowing would be

$$Q = A\sqrt{2gh} \tag{39}$$

where A is the area of the orifice.

In actual practice, the full flow Q corresponding to Eq. (25) cannot be realized. First, in a sharp-edged orifice there is a contraction of the jet taking place shortly after it leaves the opening. The section at which this contraction is complete is called the "vena contracta." The ratio of the cross section at the vena contracta, A_c, to the area of the orifice is the coefficient of contraction C_c. Therefore

$$A_c = C_c A \tag{40}$$

Values for the coefficient of contraction depend on the shape and arrangement of the orifice.

For sharp-edged orifices in thin walls, C_c is approximately 0.64 for water or other nonviscous fluids. C_c may be decreased by a cylindrical section of pipe extending inward, a so-called "Borda's mouthpiece"

(Fig. 28). Coefficient C_c of Borda's mouthpiece is $C_c = 0.5$. C_c may be increased by outward-extending mouthpieces of cylindrical or conical section.

In addition to the reduction of flow due to the contraction of the jet, there is a reduction of the theoretical velocity obtainable caused by friction and turbulence. The ratio between the actually obtainable and the theoretical velocity is called the "velocity coefficient C_V." Therefore

$$V = C_V \sqrt{2gh} \tag{41}$$

An average value for C_V is 0.97.

Both coefficients may be combined in an orifice coefficient C, so that

$$C = C_c C_V \tag{42}$$

and

$$Q = AC \sqrt{2gh} \tag{43}$$

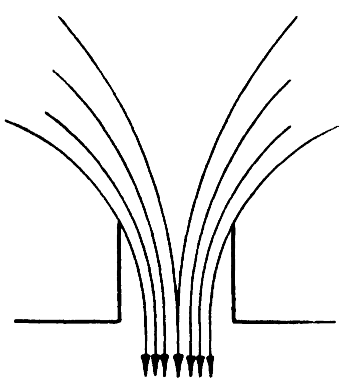


FIG. 28. Borda's mouthpiece.

For a sharp-edged orifice with $C_c = 0.64$ and $C_V = 0.97$, we have $C = 0.62$. For a nozzle with a length of about 0.6 diameter, C may be as high as 0.99.

Cylindrical outlet mouthpieces about three to five diameters in length have coefficients C from 0.82 to 0.97, depending on the radius of the inlet edge. These coefficients apply to flow of water and other fluids at Reynolds numbers above 40,000.

For flow of oil and other viscous fluids, C becomes a function of the Reynolds number. Tests made by Tuve and Sprenkle[1] indicate orifice coefficients for thin-plate sharp-edged orifices starting with 0.62 at a Reynolds number of 40,000, and increasing with decreasing Reynolds number to a maximum ranging from 0.95 to 0.68, depending upon the ratio of orifice diameter to the diameter of pipe in which the orifice was mounted. At a Reynolds number of about 40 and for all ratios, this coefficient becomes 0.65. A sharp decline takes place from then on to values as low as 0.25 to 0.30 at the extremely low Reynolds numbers.

Where the orifice diameter is small compared with the diameter of the pipe, or in cases of flow from a tank or container, orifice coefficients may be assumed to be approximately 0.65 for viscous fluids in the Reynolds number range from 40 to 40,000 for sharp-edged orifices.

10. Turbulent Flow in Noncircular Cross Sections. In the preceding chapter equations were derived for viscous flow in circular pipes and in

[1] G. L. Tuve and R. E. Sprenkle, Orifice Discharge Coefficients for Viscous Liquids, *Instruments*, November, 1933, 201.

annular and rectangular cross sections. Darcy's formula [Eq. (3)] for turbulent flow was given, which permits calculation of pressure loss in circular pipes.

In the following we shall deal with turbulent flow in noncircular sections. For this purpose we replace the term D in Darcy's formula with a new one, which is termed "hydraulic radius." The hydraulic radius of a cross section is defined as

$$r = \frac{\text{area of cross section}}{\text{wetted perimeter of cross section}} \tag{44}$$

In the case of a circular pipe

$$r = \frac{\pi D^2/4}{\pi D} = \frac{D}{4} \qquad \text{or} \qquad D = 4r \tag{45}$$

Substituting this value for D, the Reynolds number becomes

$$R = \frac{4Vr}{\nu} \tag{46}$$

and Darcy's formula becomes

$$h_f = \frac{fLV^2}{4r2g} \tag{47}$$

Expressed in commonly used terms with

L in ft
V in ft per sec
r in in.
ν in centistokes
p in psi

we have

$$R = \frac{31{,}000rV}{\nu} \tag{48}$$

$$p_f = 0.02015f\,\frac{V^2sL}{r} \tag{49}$$

In a rectangular cross section of width w and height b, the hydraulic radius r would be

$$r = \frac{wb}{2(w + b)} \tag{50}$$

and

$$p_f = \frac{0.0403fV^2sL(w + b)}{wb} \tag{51}$$

If the cross section is extremely narrow, b becomes small compared with w, and we may write

$$r = \frac{wb}{2w} = \frac{b}{2} \tag{52}$$

Then

$$p_f = \frac{0.0403fV^2sL}{b} = p_1 - p_2 \tag{53}$$

The quantity of fluid passing through a section of width w is

$$Q = 3{,}600w\sqrt{\frac{(p_1 - p_2)b^3}{fsL}} \quad \text{cu in. per min} \tag{54}$$

For an annular section of diameter D and radial width b,

$$Q = 3{,}600\pi D\sqrt{\frac{(p_1 - p_2)b^3}{fsL}} \quad \text{cu in. per min} \tag{55}$$

Coefficient f may be taken from Fig. 24 and corresponds to Darcy's coefficient for circular pipes.

It should be remembered that these formulas apply only to turbulent flow with Reynolds numbers over 2,000.

CHAPTER VII

THE GENERATION OF OIL HYDRAULIC POWER

1. ROTARY PUMPS

For the generation of oil pressure for oil hydraulic devices, the rotary type of pump is used almost exclusively. Of the different types that are in practical use, the following are the principal representatives:

Gear pumps. Used up to pressures of 1,000 psi, higher in exceptional cases. Spur, helical, or herringbone gears of the external type, and spur or helical of the internal type.

Vane pumps. Unbalanced vane pumps for low pressures, balanced vane pumps (Vickers) up to 1,000 psi, single stage.

Plunger pumps, radial and axial types.

2. GEAR PUMPS—OPERATION AND DESIGN

Since spur-gear pumps are the most generally used of the gear type of rotary pumps, description of operation and design will be based on them. Figure 29 shows the principle of a spur-gear pump.

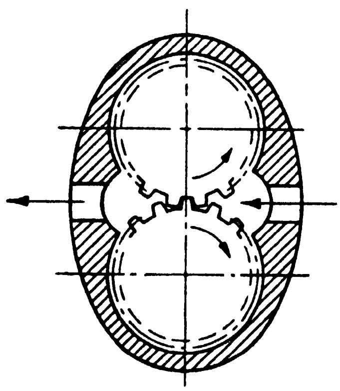


FIG. 29. Operating principle of gear pump.

Essentially the pump consists of two gears closely fitted in a housing. The oil is carried around the periphery of the revolving gears from the suction to the discharge side. The teeth are meshing between the two gears and thus prevent return of oil from the suction to the discharge.

The gear pump is of necessity a constant-displacement pump. Pressures range from a few hundred up to a thousand psi and higher. Oils of practically all viscosities encountered in hydraulic work may be used.

DISPLACEMENT AND CLEARANCE VOLUME

Figure 30, taken from an article by R. J. S. Pigott,[1] shows the two gears of a pump at the instant of first contact. A volume is enclosed between the meshing teeth that must be returned to the suction side.

[1] R. J. S. Pigott, Some Characteristics of Rotary Pumps in Aviation Service, *Trans. ASME* **66** (No. 7), 615, October, 1944.

To do this, considerable excessive pressure is generated in the tooth space, and additional loads are imposed on shafts and bearings. The total displacement of the pump is the volume carried around the periphery by both gears less the trapped volume at the point of contact.

Provisions may be made to relieve these excessive pressures and permit the trapped volume to escape back into the discharge side of the pump.

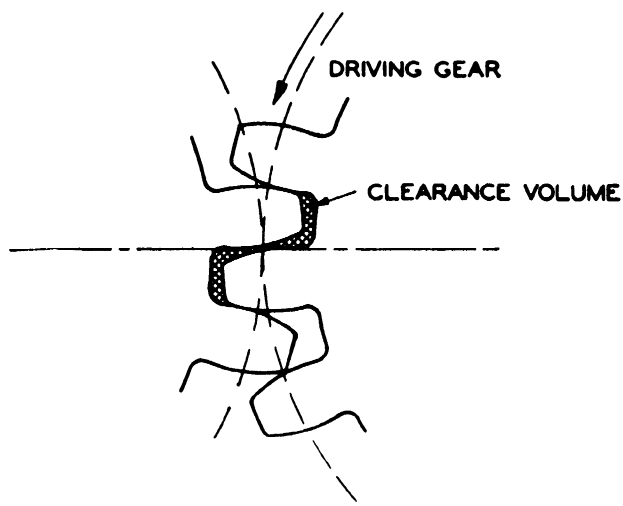


FIG. 30. Tooth contact and clearance space.

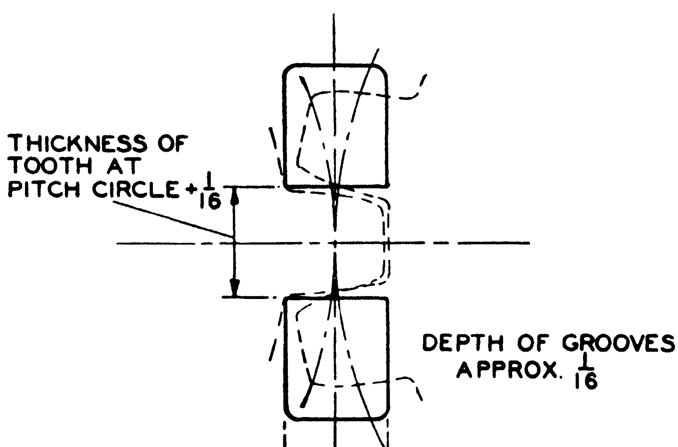


FIG. 31. Pressure-relief grooves.

This may be done by suitably arranged relief grooves, as shown in Fig. 31. These grooves are milled in the side plates or end housings.

An accurate determination of the displacement of the pump may be made by drawing the teeth in enlarged scale and planimetering the areas. Approximate displacement values may be computed by the following formula:

$$Q = \frac{[(\pi d_o^2/4) - (\pi d_i^2/4)]wn}{60} \quad \text{cu in. per sec} \qquad (1)$$

where d_o = outside diameter of gear
d_i = working-depth diameter
n = speed, rpm

The formula applies to involute gears with customarily used pressure angles. A pressure angle of 20° is recommended for pumping gears, giving more teeth in contact and less undercutting than 14½° gears. For the convenience of the reader a few of the ordinarily used formulas applied to the design of pumping gears are given in the following:

Diametral pitch = number of teeth per inch of pitch diameter $= p$

Total number of teeth $= dp = N$

Center distance = pitch diameter $= d$

Addendum $= \dfrac{1}{p}$

Working depth $= \dfrac{2}{p}$

Whole depth $= \dfrac{2.157}{p}$

Thickness of tooth $= \dfrac{1.5708}{p}$

Base-circle diameter $= d \cos \beta = d_b$

Pressure angle $= \beta$

Normal pitch $= \dfrac{\pi d_b}{N} = p_n$

$$\text{Number of teeth in contact} = \frac{\sqrt{d_o^2 - d_b^2} - d \sin \beta}{p_n} \qquad \text{for } \sqrt{d_o^2 - d_b^2} < 2d \sin \beta$$

$$= d \sin \beta \qquad \text{for } \sqrt{d_o^2 - d_b^2} > 2d \sin \beta$$

Maximum outside diameter without undercutting

$$= 2\sqrt{\left(\frac{d_b}{2}\right)^2 + (d \sin \beta)^2}$$

Speeds and Capacities

For capacities up to 20 gpm, rotative speeds of 1,800 to 1,200 rpm are very satisfactory with 8- to 5-pitch spur gears. Larger capacities may be accommodated with 3- to 4-pitch gears at 1,200 to 900 rpm.

Mechanical and Volumetric Efficiencies; Horsepower

The volumetric efficiency of the pump is the ratio between net output, as measured at a given pressure, and geometric displacement, as determined by planimetration or computation. At zero pressure, where there are no leakage losses, the volumetric efficiency is 100 per cent.

$$e_v = \frac{Q_0}{Q_g} \times 100 \qquad \text{in per cent} \tag{2}$$

The mechanical efficiency is the ratio between the theoretically required hydraulic horsepower to pump the geometric displacement and the actual horsepower, as determined by test.

$$e_m = \frac{HP_h}{HP_a} \times 100 \qquad \text{in per cent} \tag{3}$$

The hydraulic horsepower as given in Sec. 8, Chap. I, is

$$Hp_h = 0.000583Q_g p$$

where Q = displacement, gpm
p = pressure, psi

Actual horsepower may be determined by measuring the input kilowattage of the driving motor and allowing for electric-motor efficiency, which may be obtained from curves supplied by the motor manufacturer. Then

$$HP_a = \frac{\text{kilowatts} \times \text{motor efficiency}}{0.745} \tag{4}$$

The total over-all efficiency is the product of volumetric and mechanical efficiency.

$$e_t = e_v e_m \tag{5}$$

or

$$e_t = \frac{Q_0}{Q_g} \frac{HP_h}{HP_a} \tag{6}$$

and since

$$HP_h = 0.000583Q_g p$$

$$e_t = \frac{Q_0 \times 0.000583p}{HP_a} \tag{7}$$

or, in other words, total over-all efficiency is the ratio between the hydraulic horsepower of the *net* displacement and the horsepower input.

Mechanical efficiencies of spur-gear pumps actually built ranged from 80 per cent for a pump of about 10 gpm capacity to 93 per cent for a 60 gpm pump, both at 1,000 psi. At the lower pressures, mechanical efficiencies are considerably lower. Idling horsepower of spur-gear pumps may be assumed to be about 20 per cent of peak horsepower.

Volumetric efficiencies actually measured ranged from 85 to 90 per cent with oils of heavier viscosity used in hydraulic work.

The reason for the relatively low mechanical efficiency of gear pumps is the fact that these pumps are basically hydraulically unbalanced.

Referring to Fig. 28 we may see that the outlet pressure of the pump acts upon the exposed surfaces of the gears, and even beyond these by escaping through the clearances between gears and housings. This pressure has a tendency to crowd the gears over against the intake side of the housing, even where heavy shafts and bearings are used. For this reason, not all gear pumps will operate efficiently as motors. A successful attempt has been made to balance the gears hydraulically, which will be described later.

Volumetric losses in the pumps are caused by leakage in the form of viscous flow between the mating surfaces and by jet losses at the line contact of the teeth. These losses may be computed with the formulas given in preceding chapters, notably Eq. (17), Chap. V, and Eq. (43), Chap. VI.

Clearances commercially used in pumps of this type vary from 0.002 to 0.005 in. in width and diameter. In high-pressure gear pumps, provisions must be made to collect and drain off this leakage or slippage. For this purpose, the end cover plate of the pump is tapped for the attachment of a slippage drain pipe. Low-pressure gear pumps do not require this provision.

Materials

Pumps for high pressure should have hardened and ground gears. SAE 2315 or equivalent is very well suited for this purpose by carburizing and hardening. Cut gears are satisfactory for low-pressure service. Shafts are made from heat-treated alloy steel, such as 4140 or equivalent.

Roller-bearing mounting is used for high-grade high-pressure pumps. On low pressures ball or needle bearings are used, and sleeve bearings are satisfactory for light service.

Wear plates on the faces of the gears are employed for high-pressure service. These may be made from high-grade semisteel or equivalent material. Meehanite has been found very satisfactory for gear housings and end covers. Wear plates are not generally used for low-pressure pumps.

Design of a Heavy-duty Gear Pump

To illustrate the principles employed in the design of hydraulic gear pumps, a complete sample calculation will be given for a high-pressure, heavy-duty, spur-gear pump.

Example: Let us assume that a gear pump is to be designed having a capacity of 20 gpm at 1,000 psi.

We assume a mechanical efficiency of 85 per cent and a volumetric efficiency of 90 per cent. Then the gross capacity of the pump should be $20/0.9 = 22.2$ gpm, and

the input horsepower

$$HP_a = \frac{Q_0 \times 0.000583p}{e_t} = \frac{20 \times 0.000583 \times 1{,}000}{0.76} = 15.3 \text{ hp}$$

We select gears of 5-DP 20° involute system. Maintaining a proportion of pitch diameter roughly twice the width, we establish the following proportion:

$$Q_g = \frac{[(\pi d_0^2/4) - (\pi d_i^2/4)]wn}{60} = 85 \text{ cu in. per sec}$$

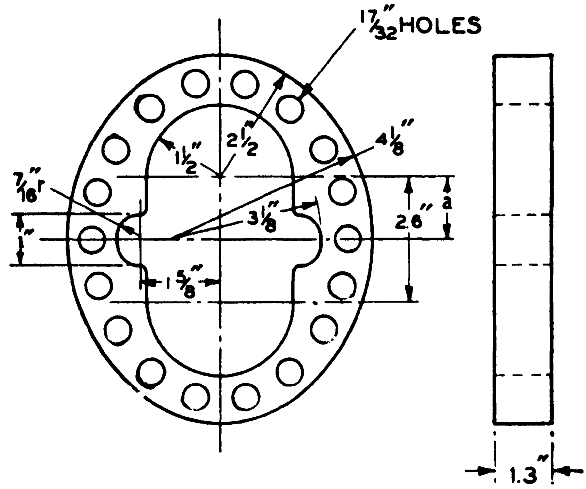


Fig. 32. Gear housing.

Assuming a pitch diameter of 2.6 in., the outside diameter will be 3 in. and the working-depth diameter, 2.2 in. With n = 1,200 rpm, Q_g will become

$$Q_g = 65.0w \quad \text{or} \quad w = 1.3 \text{ in.}$$

With a capacity of 22.2 gpm, and a velocity in the suction line of 5 ft per sec (see Sec. 8, Chap. VI), the area required for the suction passage becomes

$$\frac{22.2 \times 231}{60 \times 5 \times 12} = 1.41 \text{ sq in.}$$

To make the pump reversible, both passages should be identical. Since the gear housing in this pump is only $1\frac{5}{16}$ in. wide, the passages cannot be carried in radial direction from the housing, but must be run parallel to the axis of the gears into the rear end housing where they terminate in the pipe connections.

Figure 32 shows the layout of the gear housing with the passages. From this layout we may compute the unbalanced load as follows:

A projected area corresponding to the distance a is exposed to the full outlet pressure of 1,000 psi. The pressure then extends around the periphery of both gears to the intake passage. Owing to the fact that we have a constant leakage path around the periphery, the pressure drop must be constant. The pressure distribution on one gear would, therefore, be as shown in Fig. 33. The full pressure of 1,000 psi acts

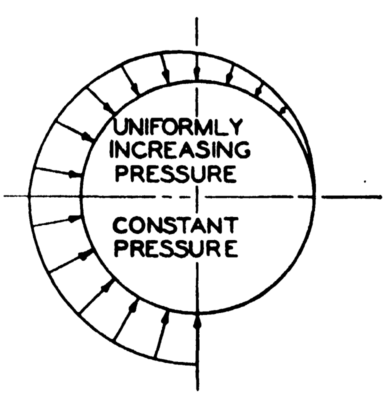


Fig. 33. Pressure distribution on gear pump.

on an arc of 90°, while a gradually decreasing pressure of 1,000 to 0 psi acts on an arc of 180°.

Referring now to Fig. 34, we have (with angles in radians)

$$p = p_{max}\frac{\alpha}{\pi} \tag{8}$$

$$\begin{aligned} dF &= prw\,d\alpha \\ &= p_{max}\frac{\alpha}{\pi} rw\,d\alpha \\ &= C_1\alpha\,d\alpha \qquad \text{for } 0 < \alpha < \pi \\ &= C_2\,d\alpha \qquad \text{for } \pi < \alpha < 1.5\pi \end{aligned} \tag{9}$$

where

$$C_1 = \frac{rwp_{max}}{\pi}$$

$$C_2 = rwp_{max}$$

We form the components in the x and y direction:

$$\left.\begin{aligned} dF_x &= C_1\alpha\cos\alpha\,dx \\ dF_y &= C_1\alpha\sin\alpha\,d\alpha \end{aligned}\right\} \qquad 0 < \alpha < \pi \tag{10}$$

$$\left.\begin{aligned} dF_x &= C_2\cos\alpha\,d\alpha \\ dF_y &= C_2\sin\alpha\,d\alpha \end{aligned}\right\} \qquad \pi < \alpha < 1.5\pi \tag{11}$$

$$F_x = C_1\int_0^{\pi}\alpha\cos\alpha\,d\alpha + C_2\int_{\pi}^{1.5\pi}\cos\alpha\,d\alpha \tag{12}$$

$$= C_1\,[\cos\alpha + \alpha\sin\alpha]_0^{\pi} + C_2\,[\sin\alpha]_{\pi}^{1.5\pi} \tag{13}$$

or

$$F_x = -(2C_1 + C_2) \tag{14}$$

$$= -rwp_{max}\left(\frac{2}{\pi} + 1\right) = 1.635rwp_{max} \tag{15}$$

A similar analysis carried out for the y component shows that the force in the direction of the y axis becomes zero. Therefore the total hydraulic load on each gear of a gear pump, designed as shown in Fig. 32, acts perpendicularly to the center line of the gears and has the magnitude

$$F_H = 1.635rwp_{max} \tag{16}$$

where

$$r = \frac{d_0}{2}$$

In our case we have

$$\begin{aligned} r &= 1.5 \text{ in.} \\ w &= 1.3 \text{ in.} \\ p &= 1{,}000 \text{ psi} \\ F_H &= 3{,}180 \text{ lb} \end{aligned}$$

To this force we must add the load produced by the tooth pressure, P. P may be computed from the torque supplied to the pump as follows:

$$P = \frac{2T}{d}$$

and

$$T = \frac{63{,}000HP}{n}$$

We have

$$HP = 15.3$$
$$n = 1{,}200 \text{ rpm}$$
$$d = 2.6 \text{ in.}$$
$$P = 620 \text{ lb}$$

The total load in the x direction, therefore, is

$$3{,}180 + 620 = 3{,}800 \text{ lb}$$

In the y direction, we have the separating force $P \tan \beta$, or with $\beta = 20°$, $620 \times 0.342 = 212$. This force may be neglected.

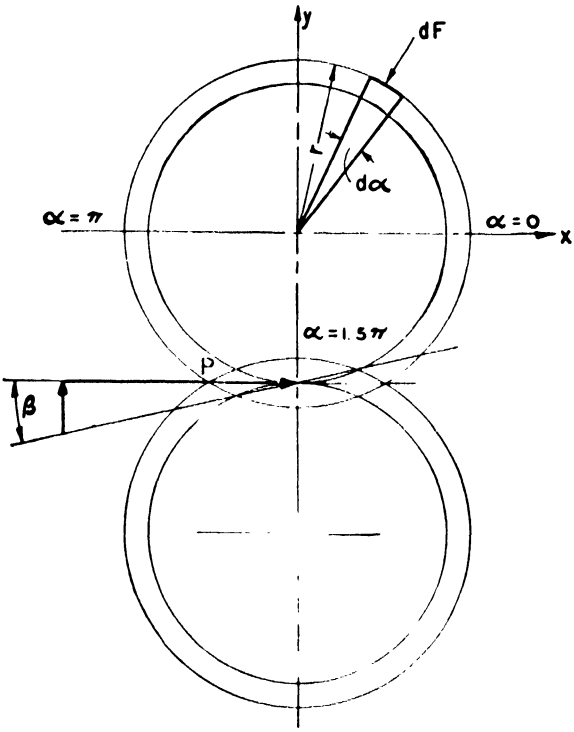


FIG. 34. Analysis of forces on gear.

We therefore have a total load of 3,800 lb that must be carried by the bearings.

Loading and speed determine selection of bearings. High-grade roller bearings are recommended for this service, although needle bearings, properly made and applied, have given satisfactory service.

The drive shaft is subjected to both bending and torsional stresses. With a gear width of 1.300 in. and faceplate thickness of ½ in., the center distance of the bearings will be 3⅛ in. With a 1¼-in. drive-shaft diameter in the center, the bending stress will be

$$\frac{1{,}900 \times (1.5625 - 0.325)}{0.1 \times 1.25^3} = 11{,}900$$

This stress will be approximately doubled owing to stress concentration caused by the keyway.[1]

[1] R. E. Peterson, Fatigue of Shafts Having Keyways, *ASTM*, *Proc.*, **32,** 413–420, 1932.

The torsional stress may be computed from the input horsepower of 15.3. The input torque will be

$$T = \frac{5{,}250 \times 15.3 \times 12}{1{,}200} \text{ in.-lb} = 810 \text{ in.-lb}$$

The torsional stress from this is

$$s_s = \frac{810}{0.2 \times 1.25^3} = 2{,}050$$

The torsional stress in the center of the drive shaft may be neglected. The ports will be carried to the rear housing through the rear wearing plate.

One-and-a quarter-inch suction-pipe connection of either flanged or screwed type is to be provided. The entire assembly is held together with eighteen ½-in.-diameter screws. Housings and wear plates are carefully surface ground or lapped so that no gaskets are required. The stress in the screws may be computed as follows: Some sections of the pump are subjected to the full working pressure, while others are at reduced pressure or suction. The strength of the bolts must be computed from the loads imposed upon them by the most heavily loaded sections. This may best be done by assuming that the entire area of the pump body is subjected to the maximum pressure and that all bolts share in the load uniformly. The internal area of the pump is approximately 15 sq in., while the land space surrounding it is approximately 18¾ sq in. Assuming a pressure of 1,000 psi for the former and 1,500 psi for the latter to prevent leakage of oil, we obtain a total force of 43,000 lb, or 2,400 lb per bolt, corresponding to a tensile stress of 16,000 psi at the root of the thread of the ½-in. bolts (NF thread).

The wear plates should be provided with relief grooves as shown in Fig. 31. With 5-DP gears, the thickness of the tooth at the pitch line is 0.315 in. The distance between the grooves therefore should be about ⅜ in. The width should be about ½ in.

Provisions should be made for leakage drain-off. Connecting holes are drilled between the bearing cup bores. The idler shaft should be drilled longitudinally to permit leakage to escape from the front bearing chambers to the slippage connection in the rear.

The shaft packing is subjected to leakage oil pressure only. Compression-type or multiple-V packing may be used for this purpose. It is very difficult to keep any of these packings from seeping. The most satisfactory results are obtained with mechanical seals such as Crane bellow seal packings. Reference to this packing will be made in Chap. VIII.

Illustrations and Description of Commercially Available Gear Pumps

The Brown and Sharpe Gear Pump. The Brown and Sharpe Mfg. Co., Providence, R.I., manufactures a complete line of gear pumps for circulating fluids, supplying lubricants and coolants, and for hydraulic work. The company's hydraulic pumps are represented by a low-pressure line for pressures up to 200 psi and high-pressure line up to 500 psi. Figures 35 and 36 show their low-pressure gear pump. These pumps are made in three sizes. The smallest has a capacity of 4.6 gpm at zero pressure

and 1,725 rpm, and the largest has a capacity of 16.4 gpm under the same conditions.

Pumps have helical gears and renewable iron bearings. They are provided with mechanical seals designed to prevent leakage and eliminate gland adjustment. Pumps may be furnished with or without integral

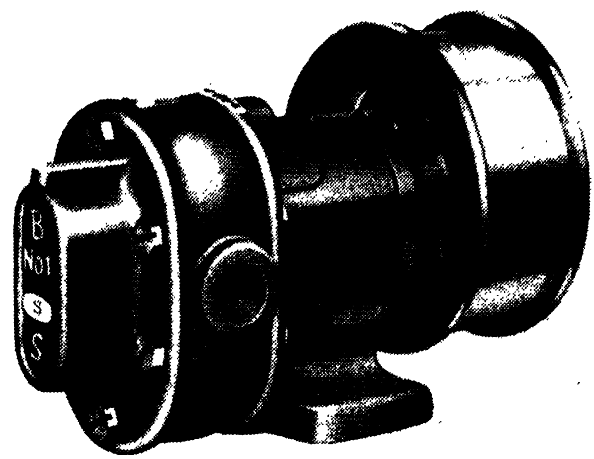


FIG. 35. Brown and Sharpe gear pump.

relief valves. Total suction lift should not extend 15 in. of mercury. A typical performance chart of a No. 2s pump is shown in Fig. 37. This chart applies to lubricating oil of 100 SSU, which the company recommends for these pumps.

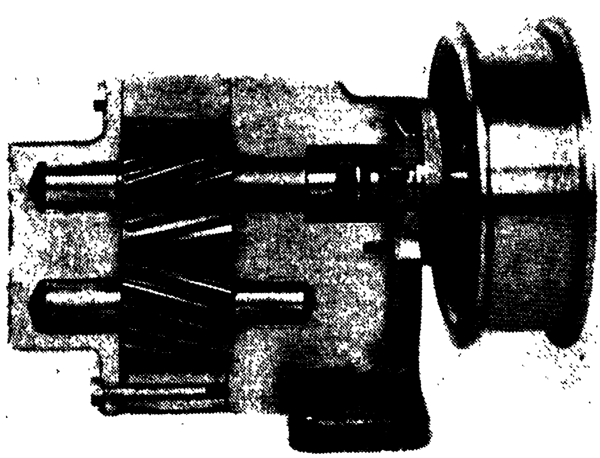


FIG. 36. Sectional view of helical-gear pump. (*Brown and Sharpe Mfg. Co., Providence, R.I.*)

The company also manufactures a larger size of low-pressure pumps, Nos. 53 and 55, having a capacity of 23.3 and 34.1 gpm, respectively, at 1,725 rpm. These pumps have roller bearings and helical gears, are made with extreme accuracy, and provide a smooth flow at high speeds. The pumps have a mechanical seal to prevent oil leakage and eliminate gland

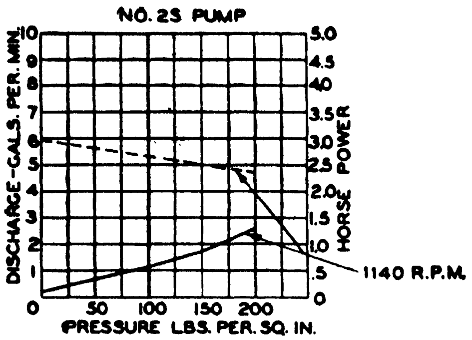


Fig. 37. Performance chart of No. 2s Brown and Sharpe pump.

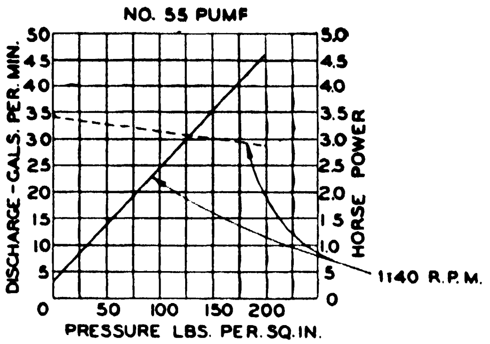


Fig. 38. Performance chart of No. 55 Brown and Sharpe pump.

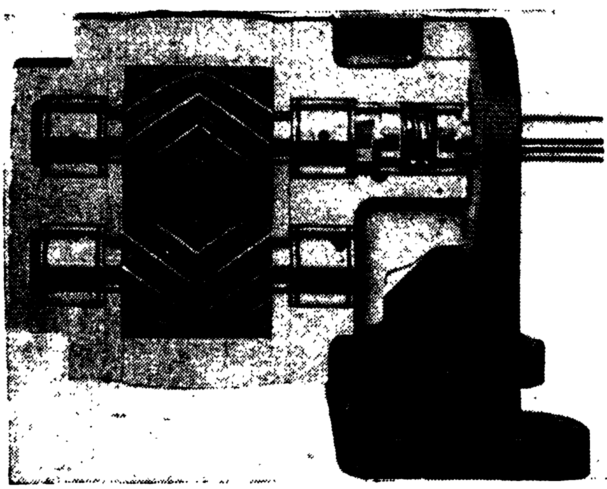


Fig. 39. Sectional view of herringbone-gear pump. (*Brown and Sharpe Mfg. Co., Providence, R.I.*)

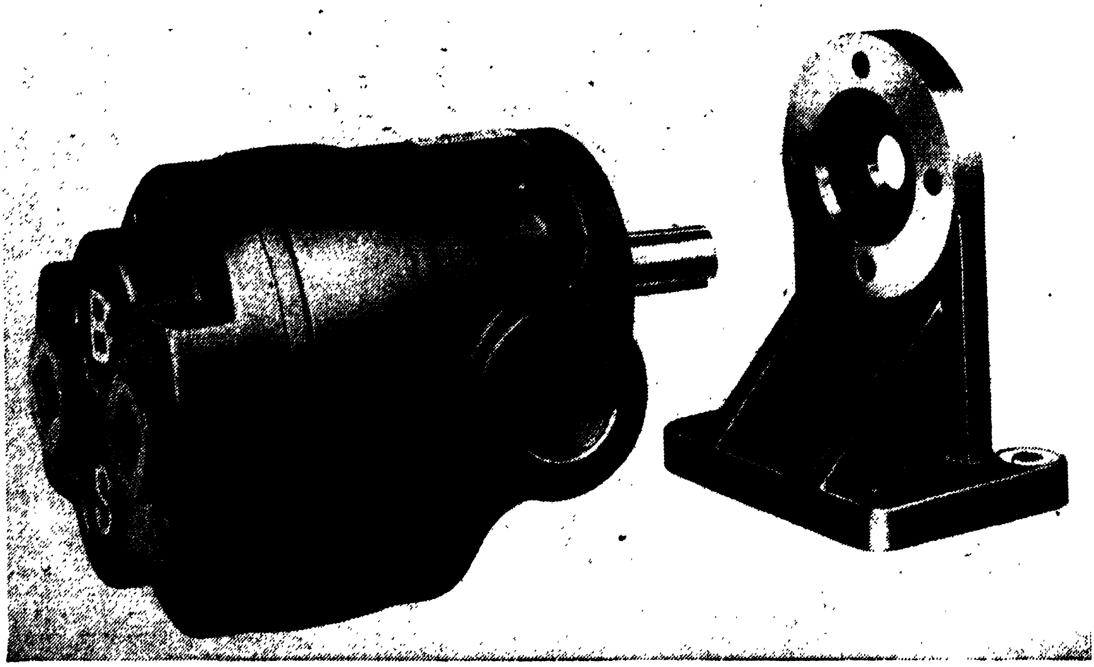


Fig. 40. Brown and Sharpe high-pressure gear pump.

adjustments. Figure 38 shows a typical performance chart of a No. 55 pump.

Figures 39 and 40 show the high-pressure pump for 500 psi. The pump is arranged for flange mounting and may be mounted on a stand as shown in Fig. 40. The pump has heat-treated steel herringbone gears, hardened shafts, and needle roller bearings. Pumps have surface-ground wear

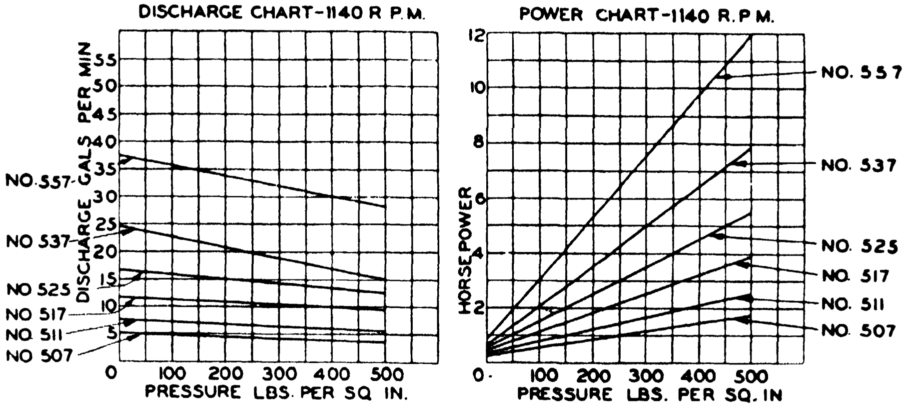


FIG. 41. Performance charts for Series 500 Brown and Sharpe high-pressure pumps.

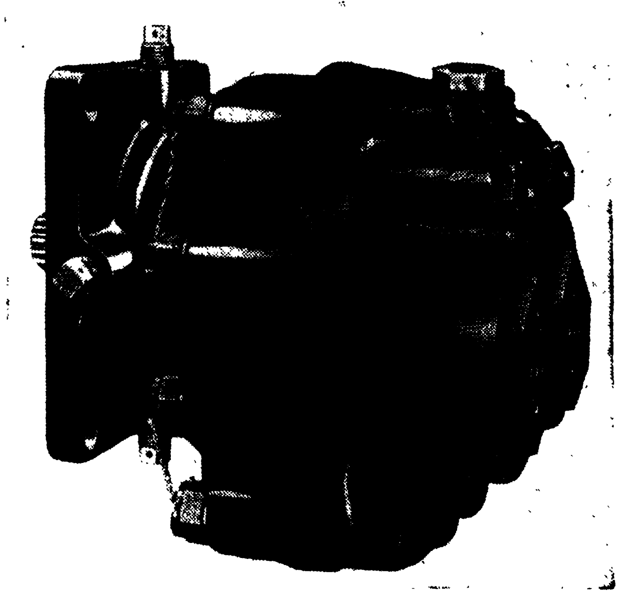


FIG. 42. Aircraft-type gear pump. (*Pesco Products Co.*)

plates, and no gaskets are required. There are six sizes ranging from 5.1- to 37.6-gpm capacity at 1,725 rpm. Figure 41 shows performance characteristics of these pumps.

The Pesco Products Co. Gear Pump. This company manufactures a line of aircraft-type high-pressure gear pumps. Five sizes are available, ranging in capacity from 0.114 to 0.555 cu in. per revolution. Speeds as high as 4,300 rpm are permissible with this pump with suitable fluids. Oil recommended is AN-VV-0-366 hydraulic fluid. Figure 42 shows their model 1-P-582-K.

Very high pressures are obtainable with this pump owing to its excellent workmanship and to a new patented principle called "pressure loading." The smallest pump operates at 1,500 psi continuously and 2,000 psi intermittently. The largest one is good for 1,500 psi continuously and 1,600 psi intermittently.

Figure 43 shows a section of the Pesco gear pump showing its working principle. Pressure loading is the patented principle used exclusively in Pesco hydraulic pumps; it utilizes the pressure developed by the pump

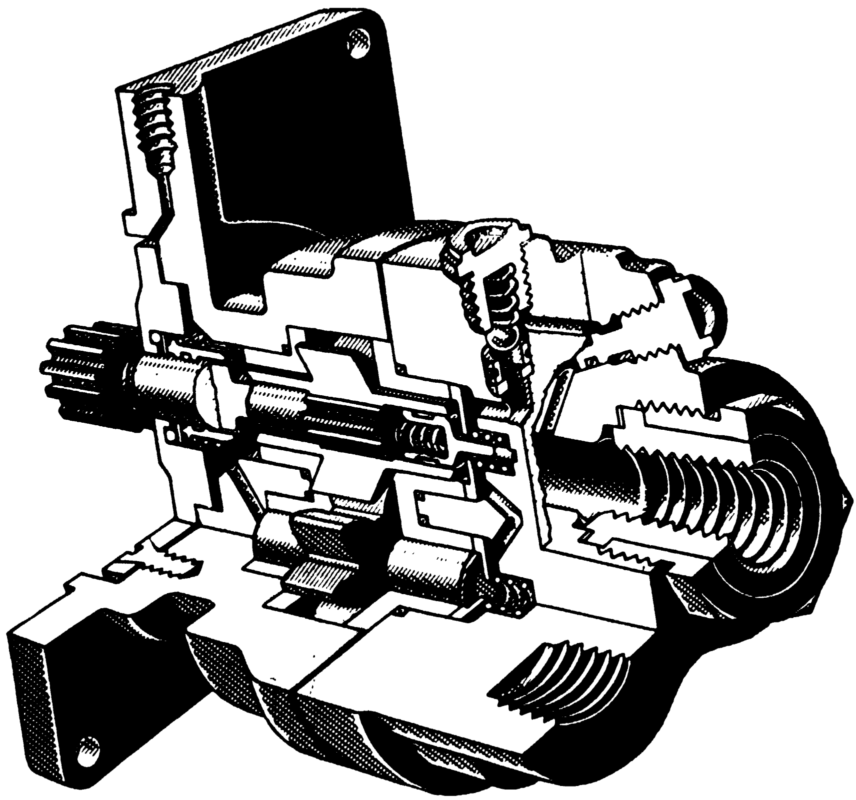


FIG. 43. Section of pressure-loaded gear pump. (*Pesco Products Co.*)

to maintain a minimum clearance between the bearings and the gears. A channel drilled in the cover conducts fluid from the discharge port to the underside of the flange on the gear-supporting-cover bearings, forcing the bearings, which are loosely fitted in the cover, against the gear faces. A similar channel carries to the pump inlet the fluid that has passed through the bearing lubricating grooves and bearing seal rings. Without increasing the size or weight of the unit, pressure loading a pump enables it to maintain a higher capacity.

Construction. The units consist essentially of an aluminum body and cover that incase two steel gears, the drive gear and the driven gear. Torque is transmitted to the drive gear, which, in turn, meshes with and drives the driven gear. The steel gears are supported on lead-bronze

bearings within the closely fitted housing. The gear shafts are hollow. Two small holes drilled in the cover connect the gear-shaft ends with the intake port to relieve the pressure at the end of the shafts. Each bearing has a flange that is held against the end of the gear it supports. The bearing also employs a seal ring to retain the fluid pressure between the bearing flange and the housing. Installation dimensions of the 1-P-582-K

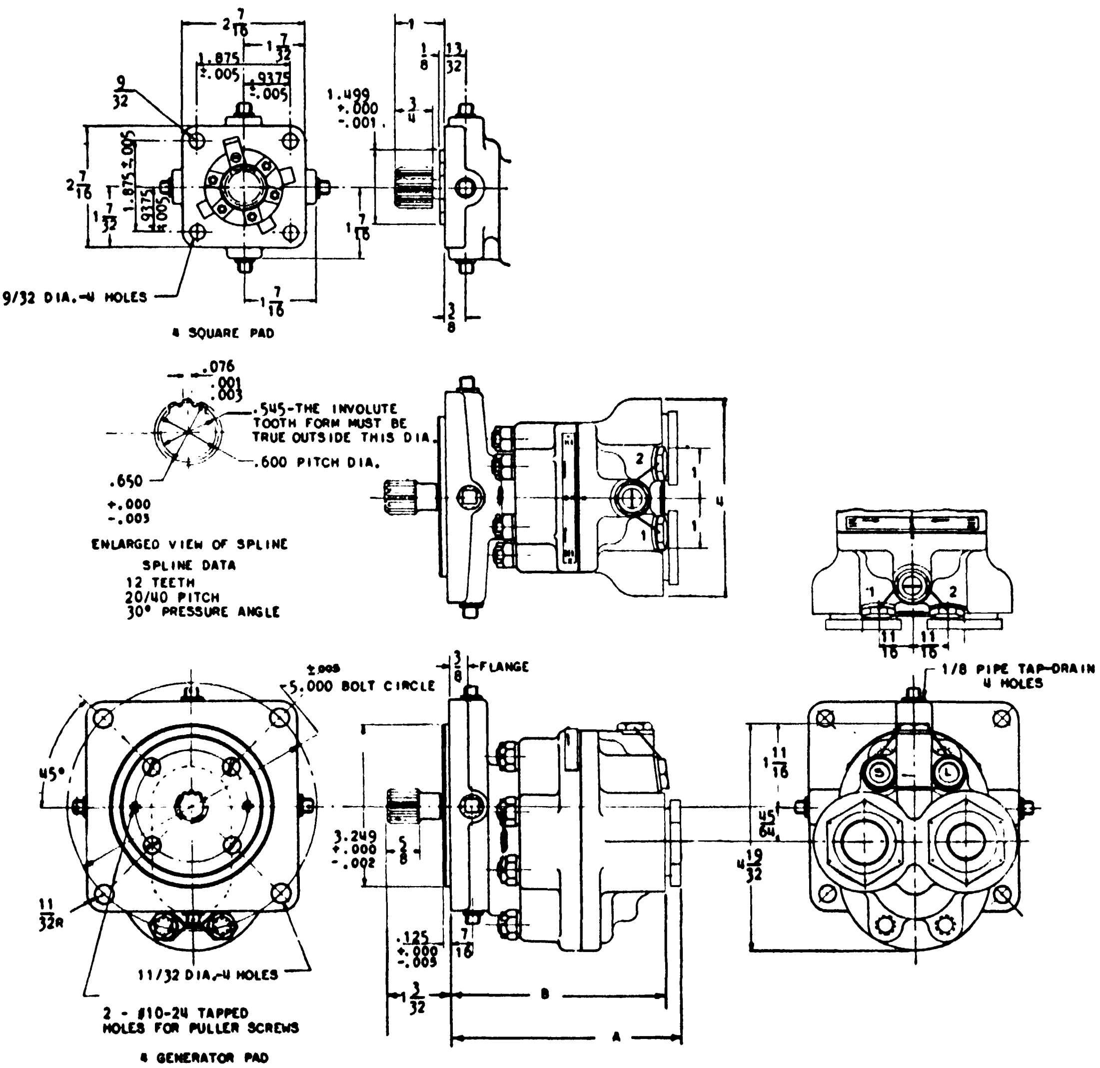


FIG. 44. Installation dimensions of 1-P-582 series pumps. (*Pesco Products Co.*)

pump, shown in Fig. 44, give an idea of the compactness and small size of this pump.

Hydro-Power Inc. Gear Pump. This company manufactures a line of high-pressure spur-gear pumps, available in six sizes ranging from 1.55 to 17 cu in. per revolution. Operating speeds range from 900 to 1,200 rpm, according to size. The pump operates with a wide range of oils with viscosities ranging from 300 to 1,000 SSU at 100°F. Figure 45 shows an exterior view of this pump. All pumps are capable of operating

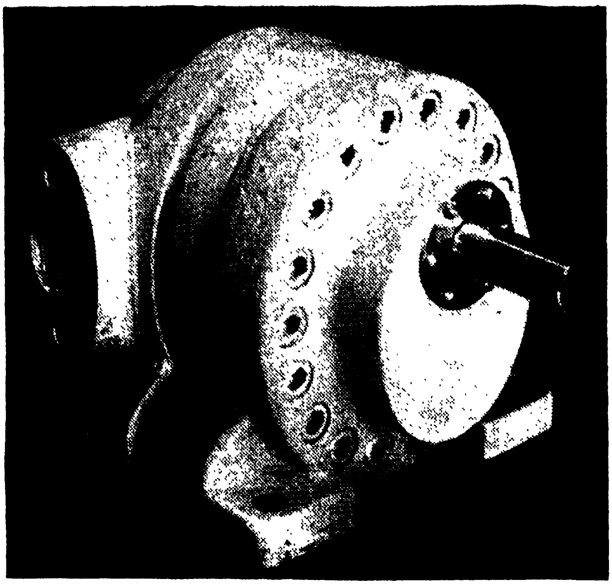


Fig. 45. Hydro-Power Inc. gear pump. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

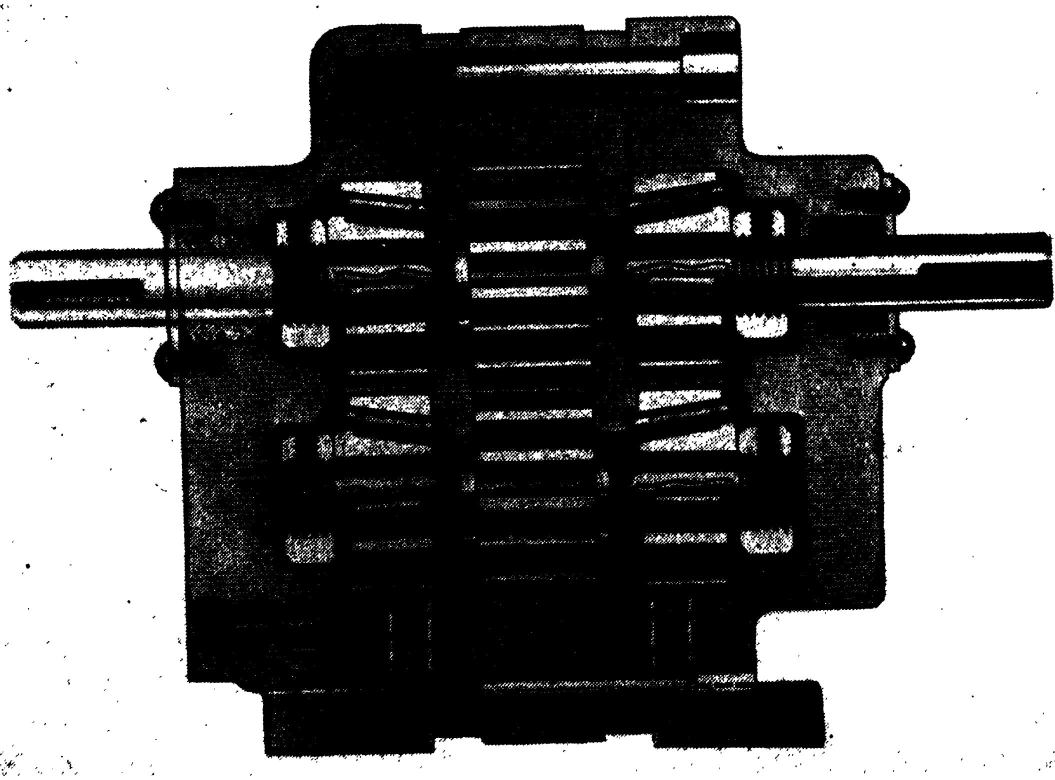


Fig. 46. Section through gear pump. (*Hydro-Power Inc.*)

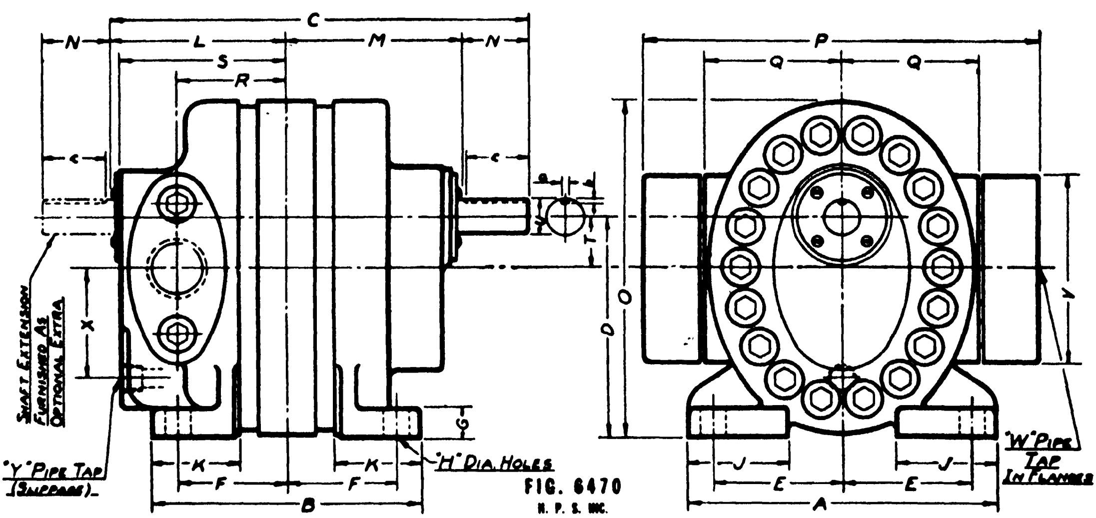
at pressures as high as 1,000 psi. Both foot- and flange-mounted models are available. A section through the pump is shown in Fig. 46.

Construction. The gears are hardened and ground chrome-nickel steel; shafts are heat-treated molybdenum steel.

Faceplates are alloy cast iron. End housings are meehanite castings, and the gear housing is a low-carbon steel forging, carburized and ground.

Long life and accurate positioning of pump gears are provided for and maintained by mounting gear shafts on precision Timken tapered roller bearings. Shaft end play is eliminated, and extremely close tolerances can be maintained between gears and housings. Wear is reduced to a minimum. Internal adjustments are provided for compensation for bearing wear.

All body joints are ground to an oiltight fit. Gaskets are eliminated, and positive minimum clearance between gears and faceplates is guaran-



MODEL "G" GEAR PUMPS — PRESSURE: 500 LBS/SQ IN CONTINUOUS: 1000 LBS/SQ IN FLASH																														
Size	GPM at 500 lbs/sq in		DIMENSIONS (INCHES)																											
	900 RPM	1200 RPM	U	Keyway a	b	c	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	V	W		X	Y
G-5	5¼	7¼	3/8	3/16	3/32	1½	7⅝	5 9/16	9 3/16	5½	3 1/16	2 3/32	¼	⅝	2½	2⅛	3 21/32	3 27/32	1⅜	8 1/16	9¼	3⅜	2 3/32	3 3/32	1¼	4¼	½	⅜	2⅛	⅜
G-10	9¼	12¼	3/8	3/16	3/32	1½	7⅝	6 3/16	9 11/16	5½	3 5/16	2 7/32	¼	⅝	2½	2⅜	4 3/32	4 3/32	1⅝	8 7/16	9¼	3⅜	2 7/32	3 7/32	1¼	4¼	½	1	2⅜	⅜
G-20	13¼	17¼	7/16	3/16	3/32	1½	7⅝	6 5/32	10 5/32	5½	3 5/16	2 41/64	¼	⅝	2½	2⅛	4 41/64	4 41/64	1⅝	8 7/16	9¼	3⅜	2 41/64	4 41/64	1¼	4¼	¾	1¼	2⅛	⅜
G-30	22	29¼	1⅛	5/16	5/32	2¼	9½	7	12¼	6¼	4	2¼	1	¾	2¼	2⅛	5⅛	4⅝	2¼	9⅜	13	4⅞	3½	5 5/16	1½	5½	1¼	1½	3	½
G-40	35	—	1⅛	5/16	5/32	2⅛	9½	7¼	13½	6¼	4	3⅛	1	¾	2¼	2⅛	5¼	5	2¼	9⅜	13	4⅞	3⅝	5 7/16	1½	5½	1¼	1½	3	½
G-60	63	—	1⅝	½	¼	3½	13½	10	16½	10¼	5⅝	3⅞	1¼	13/16	4¼	3¼	7 5/16	7 5/16	3⅜	15	16¾	6¼	4 13/16	6 13/16	2⅝	7	1½	2	5½	¾

Fig. 47. Mounting dimensions of Model G Hydro-Power Inc. pumps.

teed. The use of ground joints and closely spaced socket-head cap screws assures a strong leakproof body. Chevron-type packing is used as shaft seal. This packing is not subjected to hydraulic pressure.

Steel flanges are used to connect suction and discharge lines to pump. These flanges simplify the removal of the pump without disturbing the piping.

Operating characteristics of the Model G Hydro-Power pumps are given in Table I. Mounting dimensions of the foot-mounted line of pumps are given in Fig. 47.

The Schutte and Koerting Company's Gear Pump. This company manufactures a very complete line of gear pumps, both in spur- and herringbone-gear design. Capacities range from ⅓ to 1,000 gpm.

Standard units were originally designed for 300 psi pressure, but are suitable for operation at 500 psi, according to the manufacturer.

Castings used are meehanite B with tensile strength of 45,000 psi. Gears are heat-treated.

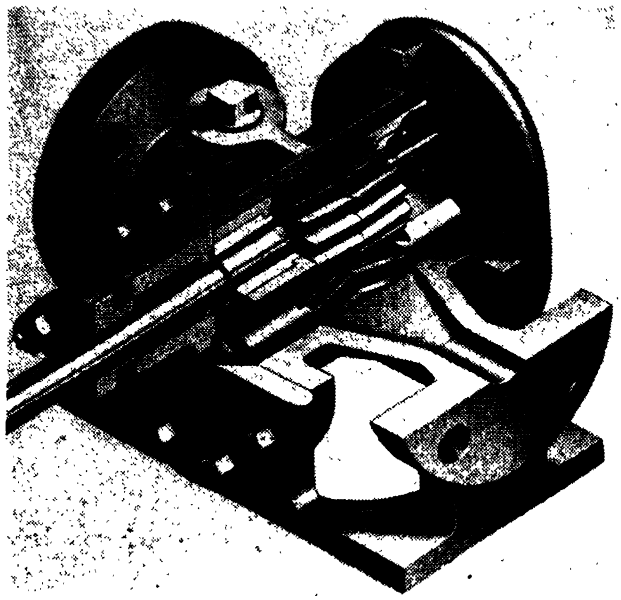


FIG. 48. Herringbone-gear pump. (*Schutte and Koerting Co., Philadelphia, Pa.*)

Spur-gear pumps are used for slow speeds of 300 to 600 rpm and have sleeve bearings. A design similar to Fig. 31 provides for relief of trapped fluid. For higher speeds up to 1,800 rpm, double helical-gear pumps are recommended. These are furnished in both sleeve- and roller-bearing design. Roller-bearing pumps are suitable for higher pressures up to 500

TABLE I. OPERATING CHARACTERISTICS OF HYDRO-POWER GEAR-TYPE PRESSURE GENERATORS, MODEL G PUMPS

Model G size	Rpm	Delivery, gpm					Horsepower input					Weight, lb	
		0 psi	250 psi	500 psi	750 psi	1,000 psi	0 psi	250 psi	500 psi	750 psi	1,000 psi	Foot	Flange
5	900	6.0	5.6	5.25	5.0	4.8	1.2	1.9	2.8	3.5	4.5	100	100
	1,200	8.0	7.5	7.25	6.75	6.5	1.5	2.5	3.75	4.75	6.0	100	100
10	900	9.6	9.4	9.25	8.9	8.7	1.4	2.9	4.2	5.7	7.0	105	105
	1,200	12.75	12.5	12.25	11.75	11.5	1.75	3.75	5.5	7.50	9.25	105	105
20	900	13.8	13.4	13.25	13.0	12.75	1.9	3.5	5.6	7.6	9.7	110	110
	1,200	18.5	18.0	17.75	17.5	17.0	2.5	4.75	7.5	10.25	13.0	110	110
30	900	24.0	23.0	22.0	20.8	20.0	2.25	5.1	8.5	12.0	16.2	160	160
	1,200	32.0	30.5	29.25	27.75	26.5	3.0	6.75	11.25	16.0	21.5	160	160
40	900	38.0	36.5	35.0	33.5	32.0	3.0	8.5	14.0	19.5	25.0	175	175
60	900	66	64.5	63	61	59	9.0	16.5	25	33.5	42	400	400

psi. Oils of all viscosities encountered in hydraulic work are suitable for use with these pumps.

A section of the herringbone pump is shown in Fig. 48. Both gears and sleeve bearings, forming the faceplates for the gears, are mounted in the housing bore, eliminating overhang of bearings and reducing shaft deflection. Relief grooves of conventional design may be seen cut into the faces of the bearings in Fig. 48 (see also Fig. 30).

The pumps are well adapted for prefill service at moderate pressure for rapid advance of hydraulic rams. Pressures of from 250 to 300 psi are generally used for this purpose, and large capacities are often required.

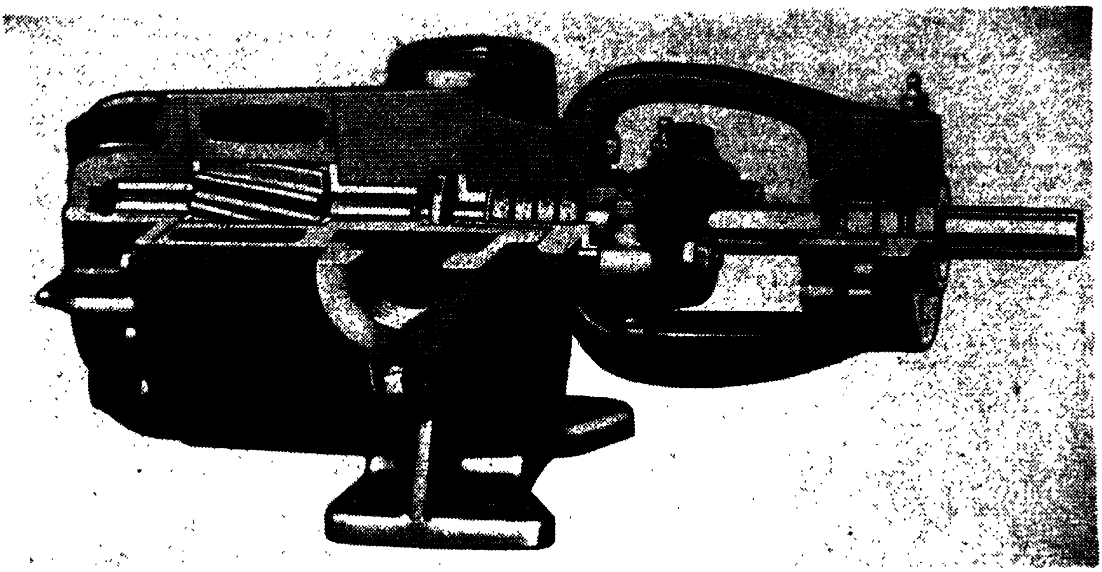


FIG. 49. Helical-gear pump. (*Geo. D. Roper Corporation, Rockford, Ill.*)

Over-all efficiencies at 300 to 500 psi range from 45 to 70 per cent depending on oil viscosity, type of bearings, and gears.

Gear Pumps of Geo. D. Roper Corporation. This company makes several lines of gear pumps for both hydraulic and circulating-fluid service. In the following we shall deal with their hydraulic pumps only.

For pressures up to 300 psi, they build a helical-gear pump with sleeve bearings. The design of this pump is shown in Fig. 49. Pumps may be equipped with packings and outboard bearing on drive shaft as shown in Fig. 49, or with mechanical shaft seals. The driving gear is driven from the drive shaft by an Oldham coupling so that misalignment stresses cannot be transmitted to the gears, which are in axial hydraulic balance. Flanges on the bearings form the wear plates for the gear faces in this design. Oil used in performance test has 300 SSU viscosity. Rated capacities run from 1 to 300 gpm. Pumps show about 70 per cent over-all efficiencies at 300 psi. Idling horsepowers (50 psi pressure) range from 0.17 for a 1-gpm pump to 15 hp for a 300-gpm pump.

The company also builds a high-pressure line of gear pumps ranging in capacity from 5 to 75 gpm. The high-pressure pump is a spur-gear pump with gears mounted on needle roller bearings and with brass wear plates. The pump features a splined drive shaft entering a spline in the driving gear. Thus gears are protected from misalignment and driving loads. The company is one of the pioneers in the design and manufacture of high-pressure gear pumps. The pump is shown in Fig. 50.

Pumps operate at speeds ranging from 1,200 rpm for the smaller sizes to 900 rpm for the larger. Over-all efficiencies at 1,000 psi range from

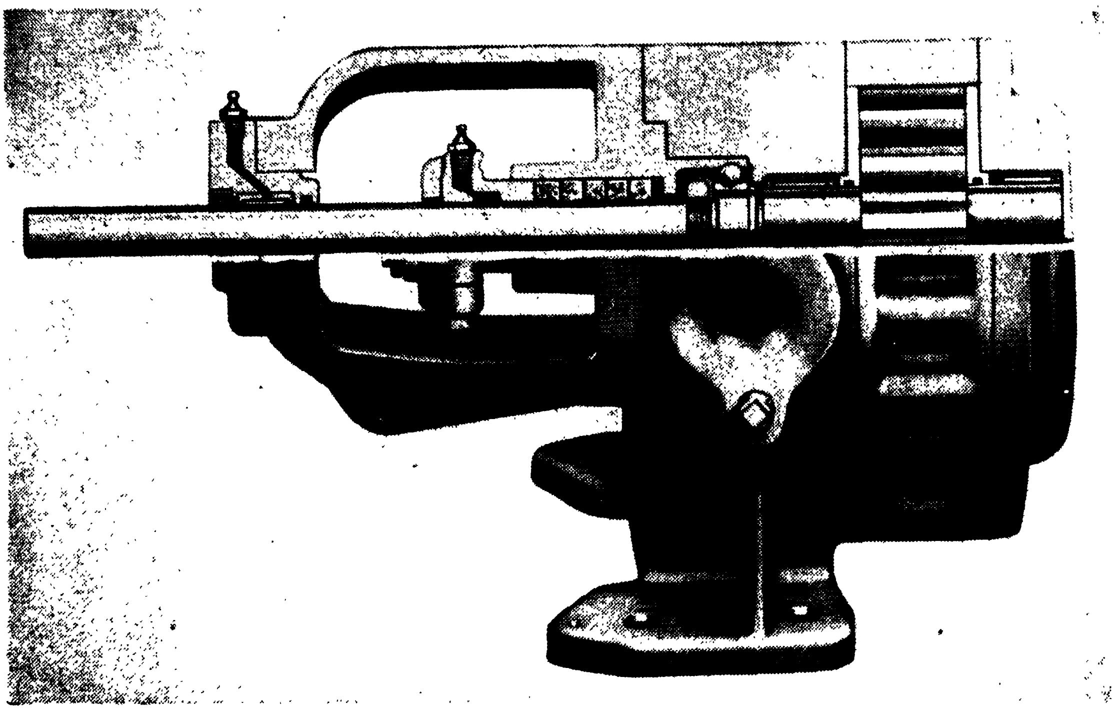


FIG. 50. Spur-gear pump. (*Geo. D. Roper Corporation, Rockford, Ill.*)

65 to 75 per cent, according to size. Oil of practically all viscosities used in hydraulic work may be accommodated.

The Vickers Gear-type Fluid Motor. Vickers Inc., Detroit, features a balanced gear pump that will operate as a motor. Ordinary gear pumps will not operate satisfactorily as motors, as the pressure applied to them will have a tendency to lock the gears against the housing. In the Vickers gear motor, pressure is conducted by passages from the intake and pressure ports to suitable pockets diametrically opposite, so that the gears are in radial hydraulic balance. A section of the Vickers gear motor is shown in Fig. 51.

Motors operate at 70 to 80 per cent over-all efficiency and develop up to 90 per cent of their theoretical torque rating. Maximum pressures are 1,000 psi, and the units are recommended for pressures of 600 to 800 psi. Recommended oil viscosities are from 225 to 315 SSU at 100°F. Motors

are furnished in both flanged and foot-mounted models. The motors are rated in sizes from 15 to 90 in.-lb per 100 psi.

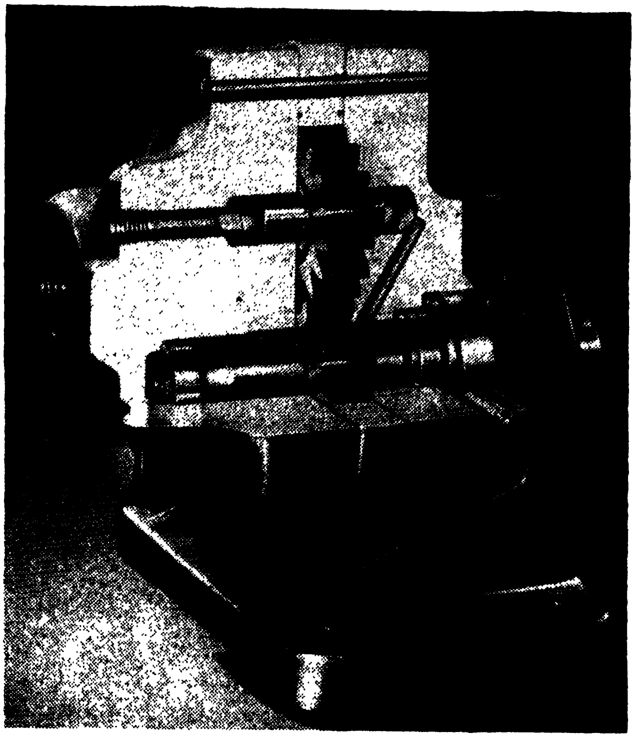


FIG. 51. Balanced gear motor. (*Vickers Inc., Detroit, Mich.*)

INTERNAL-GEAR PUMPS

The operating principle of an internal-type-gear pump is very similar to that of an external-gear pump. Several patented designs have made their appearance on the market and will be shown and described in the following.

The Sundstrand Machine Tool Co. This company manufactures a pump under the trade name "Rota Roll." As may be seen from Fig. 52, the pump consists of a pinion gear, known as the "roller," driving an internal gear known as the "rotor." Oil is carried by both gears around their periphery from suction to discharge, and a rolling seal is provided at the contact point between the two pumping members. To prevent trapping, the outlet port of the pump is carried up to the contact point so that all

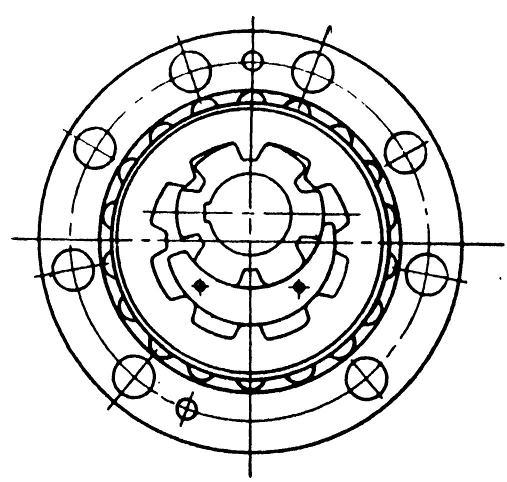


FIG. 52. Principle of Rota Roll pump. (*Sundstrand Machine Tool Co., Rockford, Ill.*)

oil is forced into the outlet. A crescent-shaped spacer, typical for most internal-type-gear pumps, serves as separator between intake and outlet openings.

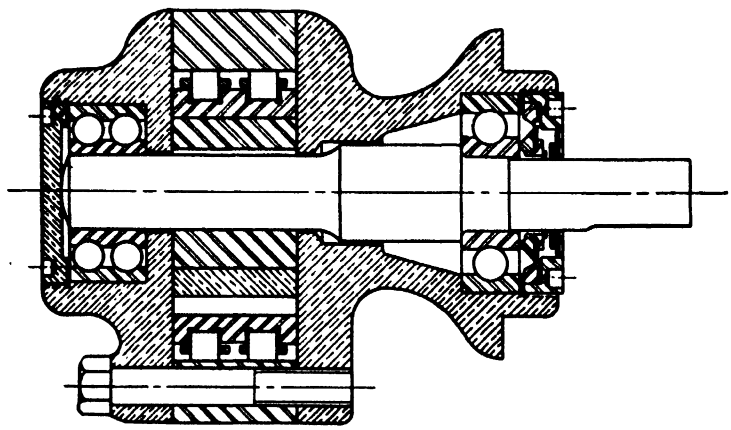


FIG. 53. Section of Rota Roll pump. (*Sundstrand Machine Tool Co., Rockford, Ill.*)

The rotor is mounted on roller bearings, and the drive shaft carrying the roller operates on ball bearings. A sectional view of the Sundstrand pump is shown in Fig. 53.

Pumps are made in capacities ranging from 2 to 13½ gpm and operate at maximum speeds of 1,500 rpm. Efficiencies at 1,000 psi are 70 to 77 per cent, according to size. Oil recommended is 150 SSU at 100°F. An exterior view of the pump is shown in Fig. 54.

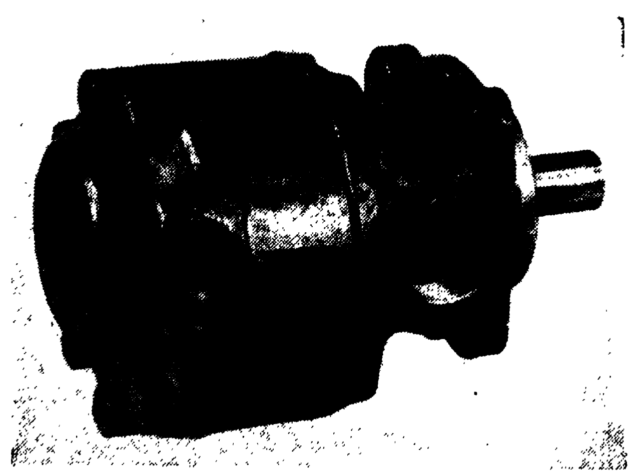


FIG. 54. Sundstrand Rota Roll pump. (*Sundstrand Machine Tool Co.*)

Gulf Research and Development Co. External-internal-gear pumps with teeth of standard involute design are restricted to combinations where the pinion has two or more less teeth than the ring gear. Pinion and ring gear are separated by a crescent-shaped divider. An interesting design

employing logarithmic spirals as tooth curves developed by R. J. S. Pigott of the Gulf Research and Development Co. is shown in Fig. 55.

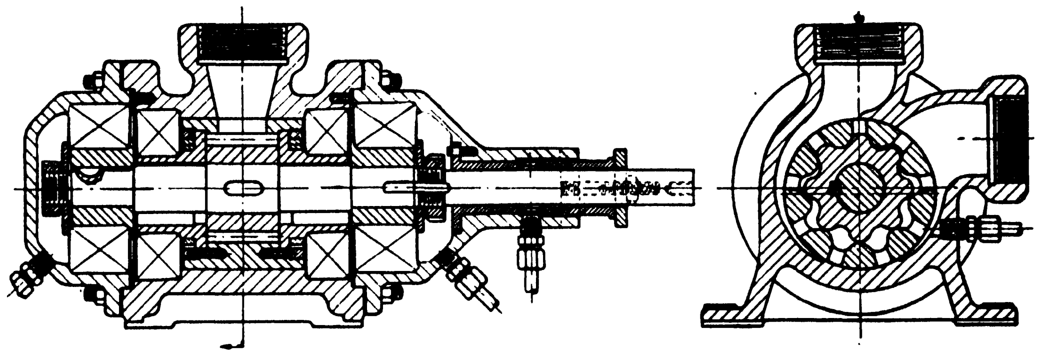


FIG. 55. Pigott internal-gear pump. (*Gulf Research and Development Co., Pittsburgh, Pa.*)

The pinion in this pump has one less tooth than the ring gear. The curves of the teeth in both gears are logarithmic spirals represented by the equation, in polar coordinates,

$$r = Ce^{YA} \tag{17}$$

where r = radius vector
A = angle of radius vector
C = factor equal to one-fifth the tooth height
Y = constant, which equals approximately 0.7 for the teeth and 0.8 for the tooth spaces of the rack from which the external gear and the cutter for the internal gear are made.

The radius vector of the logarithmic spiral has a constant angle with the tangent on the curve. From Eq. (17) we have

$$\log_e r = YA + \log_e C \tag{18}$$

or

$$A = \frac{\log_e r - \log_e C}{Y}$$

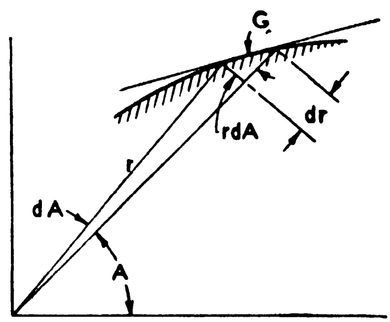


FIG. 56. Determination of tangent angle.

From Fig. 56

$$\tan G = \frac{r\,dA}{dr} \tag{19}$$

$$\frac{dA}{dr} = \frac{1}{rY} \tag{20}$$

$$\frac{r\,dA}{dr} = \frac{1}{Y} = \tan G \tag{21}$$

Pitch diameters and pitch of gears must be so chosen that the difference in pitch diameters is equal to the tooth height and that the number of

teeth in the ring gear equals the number of teeth in the pinion plus one. For this relationship Pigott[1] gives the values shown in Table II.

TABLE II

Pinion			Ring gear		
Pitch diam.	No. of teeth	Tooth height	Pitch diam.	No. of teeth	Tooth height
3.00	6	0.50	3.50	7	0.50
3.50	7	0.50	4.00	8	0.50
4.00	8	0.50	4.50	9	0.50
4.50	9	0.50	5.00	10	0.50

Pinion and gear operate together with smooth continuous action and at uniform angular velocity. The conventional crescent is eliminated. Pinion and ring gear are mounted on ball bearings. Intake and outlet of oil to and from the tooth spaces is accomplished by slots in the ring gear,

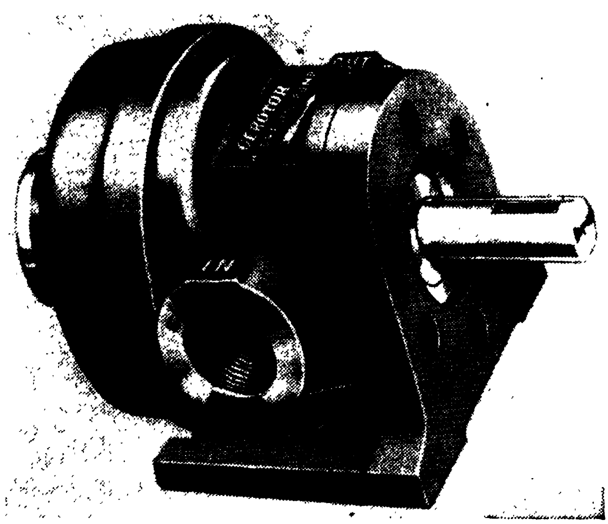


FIG. 57. Gerotor rotary pump. *(Gerotor May Corp., Baltimore, Md.)*

as shown in Fig. 55. The pump is designed for a working pressure of 600 psi.

Gerotor May Corporation. This company manufactures a very complete line of rotary hydraulic pumps of the internal-gear type. Three models are offered, known as Type B, Type H, and Type 0. Three to four capacities are available in each type, covering a wide range to fill almost any requirement.

Figure 57 shows the Gerotor pump in a foot-mounted model. Flange mounting is also available. The pump has two rotors, the pinion or

[1] U.S. Patent 2,225,515.

inner rotor having one tooth less than the ring gear or outer rotor. The conventional crescent is again eliminated. The pump operates in the same manner as the Pigott pump, each tooth of the pinion being in continuous sliding contact with a tooth on the ring gear, providing fluidtight engagement. The pump is valved by crescent-shaped slots extending

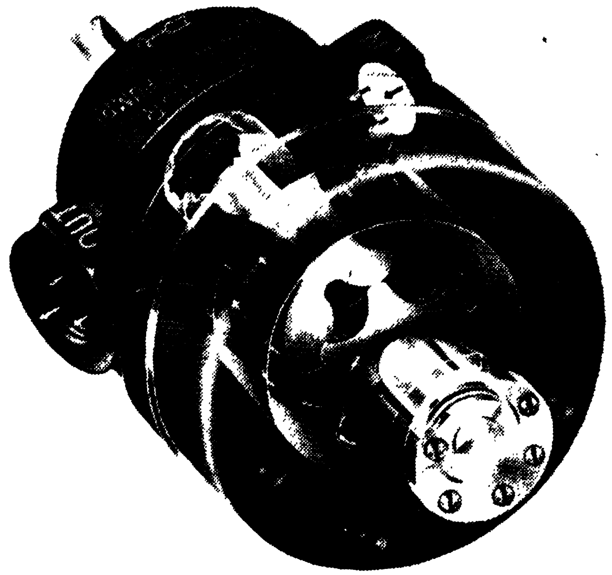


FIG. 58. Gerotor pump. (*Gerotor May Corp., Baltimore, Md.*)

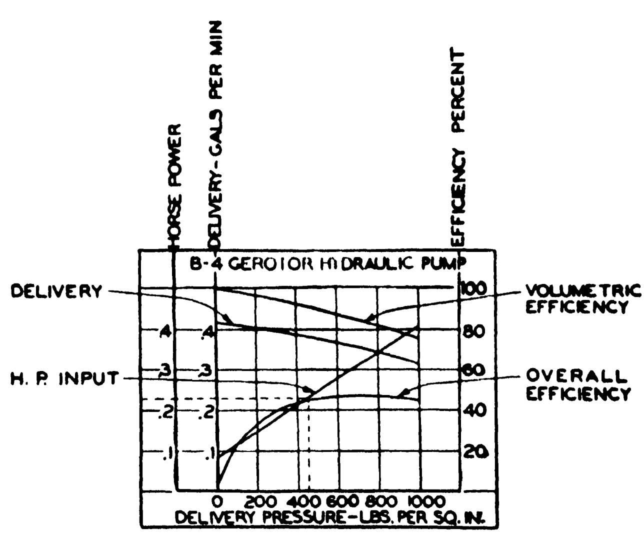


FIG. 59. Performance curves for B.4 Gerotor pump. (*Gerotor May Corp., Baltimore, Md.*)

axially. Holes are drilled in the tooth spaces of the outer rotor, permitting pressure to escape to the outside area and balance the outer rotor. The inner rotor is mounted on the drive shaft and carried in plain bearings. Maximum pressure recommended for these pumps is 1,000 psi. Oil with viscosities of from 140 to 200 SSU at 120°F are recommended. A phantom view of the Gerotor pump is shown in Fig. 58.

Sizes and Capacities. Type B Gerotor pumps vary in capacity from about ⅓ to 1½ gpm. Efficiency of these small-capacity pumps is naturally low. Figure 59 shows typical performance curves for a B.4 unit.

Type H pumps run from 3 to 12 gpm, Type 0 from 17 to 30 gpm. Performance curves for an 0-25 unit are shown in Fig. 60.

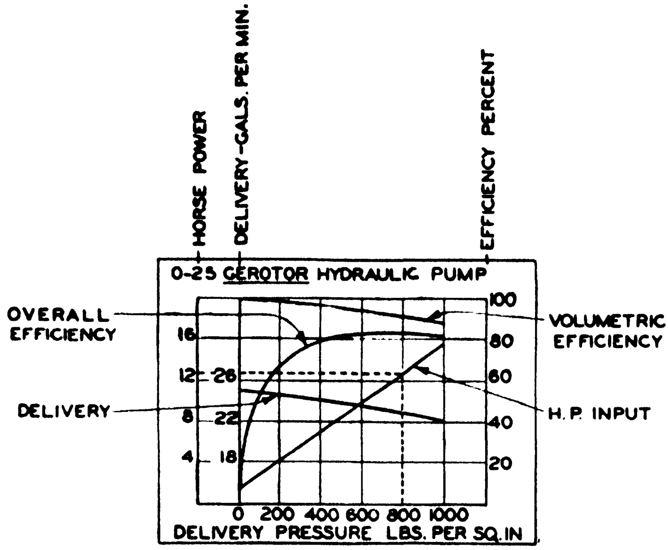


FIG. 60. Performance curves for 0-25 Gerotor pump. (*Gerotor May Corp., Baltimore, Md.*)

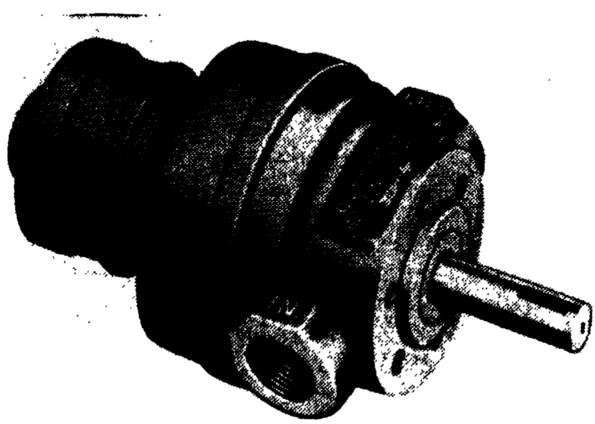


FIG. 61. Gerotor double-unit hydraulic pump. (*Gerotor May Corp., Baltimore, Md.*)

Any of the sizes may be combined into double pump units. This offers a simple and compact means of supplying two circuits with different and independent pressure and volume requirements. A double pump unit is shown in Fig. 61.

3. VANE PUMPS—OPERATION AND DESIGN

The operating principle of a vane pump is shown in Fig. 62. The pump consists of a rotor mounted in a housing, generally between closely fitted end plates. The rotor is provided with slots in which vanes or blades are

mounted. The rotor revolves on a shaft mounted eccentrically in respect to the housing, and the vanes slide on a hardened track inside the housing. As the vanes proceed from the point of closest contact between rotor and housing, increased space will be created between rotor and housing. This space fills with oil supplied through a peripheral valve slot from the

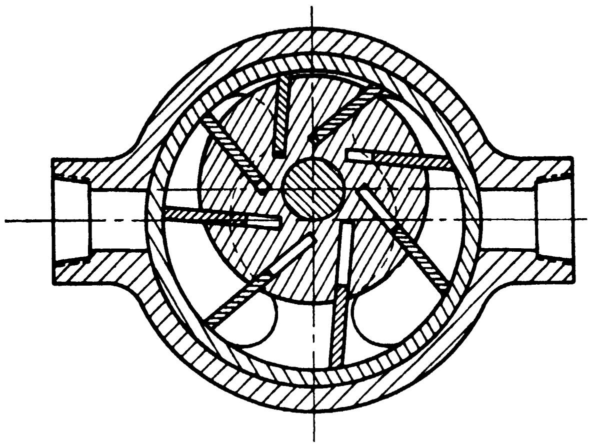


FIG. 62. Vane pump.

suction of the pump. After the point of maximum distance between rotor and housing has been passed, the spaces between the vanes again decrease, and oil is discharged through the opposite valve port into the exhaust of the pump.

Pumps of a design as shown in Fig. 62 are subject to heavy unbalanced loads. A successful design of a balanced vane pump is the well-known Vickers vane pump. This pump has two intakes and two outlets diametrically opposite each other; thus the hydraulic loads on the rotor balance each other. A more detailed description of this pump will be given later.

Vane pumps are being made in both constant- and variable-delivery design.

VANE-PUMP DISPLACEMENT

The vane pumps are, by nature of their design, well adapted to displace large volumes of oil in comparatively small space. The displacement of a vane pump may be computed in the following manner:

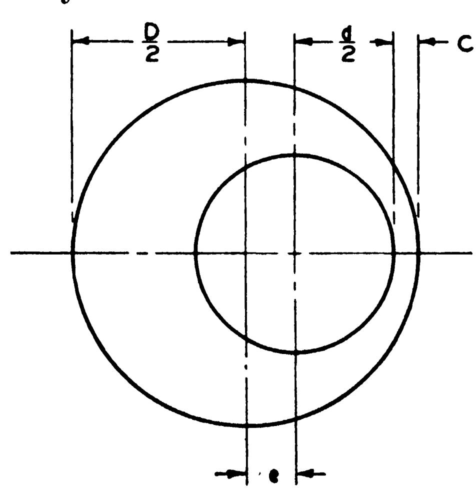


FIG. 63. Rotor and housing.

Figure 63 shows the arrangement of rotor and housing. D represents the diameter of the housing and d the diameter of the rotor. e is eccentricity between rotor and housing. If we neglect the thickness of the

vanes, then the volume carried by them from suction to discharge in one revolution must be

$$V \text{ per in. of width } = \left(\frac{D}{2} + e\right)^2 \pi - \left(\frac{d}{2}\right)^2 \pi - \left[\left(\frac{D}{2} - e\right)^2 \pi - \left(\frac{d}{2}\right)^2 \pi\right]$$

$$V = 2\pi De$$

Therefore

$$Q = \frac{2\pi Denw}{60} \qquad \text{cu in. per sec} \qquad (22)$$

where w is the width of the vanes or rotor.

Similarly in a balanced vane pump with a cam with a major axis of $D + 2e$ and a minor axis of $D - 2e$, the total capacity will be

$$Q = \frac{4\pi Denw}{60} \qquad (23)$$

Efficiencies of vane pumps may be computed by the use of Eqs. (2) to (7) given in Sec. 2.

Excellent efficiencies are obtainable with properly designed vane pumps, but first-class workmanship and a high degree of accuracy are necessary. Vickers balanced vane pumps show volumetric efficiencies of over 90 per cent and total efficiencies well above 80 per cent over a wide range of pressures.

Materials and Design

Rotors should be made of alloy steel carburized and hardened, such as SAE 2315, 3120, or 4615. Drive shafts, if not integral with rotor, may be made of medium-carbon chrome-nickel or molybdenum steel. Vanes should be made of a steel containing tungsten, as at the high rubbing speed considerable heat is generated, which will soon draw the temper on any ordinary alloy steel. High-speed tool steel properly hardened has been found excellent for this purpose. The vane track or pressure chamber may be made from a carburizing grade of alloy steel and hardened. End plates are made from manganese or silicon bronze. Meehanite or high-test cast iron will serve well for housings or end covers. Unbalanced vane pumps require heavy shafts and sturdy roller bearings to take the hydraulic loads. In a balanced pump plain bearings suffice. Regular models of vane pumps are made with splined shafts, engaging splined bores in rotor to prevent transmission of misalignment stresses to the rotor.

It is important to keep in mind that pressure must be supplied to the bottom of the vane slots when vanes pass the pressure area and cross the lap spaces, as the centrifugal force of the vanes is insufficient to permit

generation of pressure. This may be done in a number of ways. The Vickers pump employs side plates, counterbored at the bottom of the vane slots, and passages leading from the pressure side to this counterbore to supply a permanent and continuous pressure against the bottoms of the vanes, forcing them out against the track to maintain a seal.

On the Racine pump, slots are milled into the side plates connected to the pressure side and extend past the lap spaces on both sides to maintain pressure against the vanes when they are traveling on the pressure side. On the suction arc, a corresponding slot connects the bottom of the vanes to the suction side. Thus the vanes are balanced at practically every point of their travel and the vane displacement is utilized in the displacement of the pump. Balance has been accomplished in the "Dudco" pump (Detroit Universal Duplicator) by the provision of two vanes in one vane slot, mounted back to back with beveled edges. This permits the pressure at the bottom of two vanes to reach the space at the top and practically balance the vanes.

To illustrate the principles employed in the design of vane pumps, a sample calculation will be given in the following for a vane pump of unbalanced design and constant delivery.

Example: Let us assume that a vane pump is to be designed for a capacity of 20 gpm at 1,000 psi.

With an estimated volumetric efficiency of 90 per cent and a mechanical efficiency of 90 per cent, we have

$$Q_a = \frac{Q_0}{0.90} = 22.2 \text{ gpm} = \frac{22.2 \times 231}{60} = 85 \text{ cu in. per sec}$$

$$HP_a = \frac{Q_0 \times 0.000583p}{e_t} = \frac{20 \times 0.000583 \times 1{,}000}{0.81} = 14.5 \text{ hp}$$

With $n = 1{,}200$, $Q = 85$. We select an eccentricity $e = 5/32$ in. Then

$$Dw = \frac{85 \times 60 \times 32}{1{,}200 \times 2\pi \times 5} = 4.34 \qquad \text{[from Eq. (22)]}$$

$$D = 3.3 \text{ (assumed)}$$

$$w = 1.31$$

$$\frac{d}{2} = \frac{D}{2} - (e + c) \qquad \text{(see Fig. 63)}$$

Assuming $c = 0.015$, and since $e = 0.15625$,

$$\frac{d}{2} = 1.65 - 0.17125 = 1.479$$

$$d = 2.958$$

$$\text{Total pressure on rotor} = 1{,}000Dw = 4.340 \text{ lb}$$

$$\text{Bending moment on shaft} = 2.170(1.625 - 0.327) = 2.800$$

With a $1\frac{1}{4}$-in. shaft we have a stress of

$$\frac{2.800}{0.195} = 14.150 \text{ psi}$$ [1]

For the design of the pump, we assume the following:

$$\text{Number of vanes} = 16$$
$$\text{Thickness of vanes} = \tfrac{3}{32}$$

The vane slots are slanted to accommodate longer vanes.

We next compute the bending moment on the vanes:

$$M = \frac{p \times 0.33 \times 0.33}{2} \text{ in.-lb per in.} = 54 \text{ in.-lb per in. of width}$$

$$\text{Section modulus} = \frac{(3/32)^2}{6} \text{ cu in. per in.} = 0.00146 \text{ cu in. per in.}$$

$$\text{Bending stress} = \frac{54}{0.00146} = 37{,}000 \text{ psi}$$

It may be seen readily that the eccentricity of a vane pump is limited by the bending strength of the vanes. At the same time the thickness of the vanes must be held down to avoid excessive pressures on the contact line between vane and pressure ring when hydraulic pressure is applied to hold the vane out against the pressure ring.

The lap space between the ports should be somewhat wider than the pitch of the vanes to prevent short circuiting between suction and discharge ports. The pitch distance between vanes is

$$\frac{\pi d}{16} = \frac{\pi \times 2.958}{16} = 0.58 \text{ in.}$$

Lap space should be about $\frac{3}{4}$ in., port slot width $\frac{5}{16}$ in.

The design results in the dimensions shown in Fig. 64. A collector groove is shown connecting the bottoms of the vane slots. Pressure is supplied to this collector groove from the pressure side of the pump to hold the vanes out against the pressure ring on pressure arc and lap spaces. A similar groove connects the bottoms to the suction port on the suction arc. Care must be taken to make the passages to the collector rings of ample capacity to ensure ample oil supply on suction stroke and prevent the pressure under the vanes from leaking out faster than it is supplied, which would make the pump inoperative. Milled slots serve the purpose in this design.

Shaft Bearings. The bearings must support the hydraulic load of 4,340 or 2,170 lb each. At the drive-shaft end, a double-row ball bearing, medium-type SKF No. 5,306, is provided. This bearing carries radial load and provides axial location. This bearing at a load of 2,170 lb and a speed of 1,150 rpm will have a minimum life expectancy of 650 hr. A Hyatt No. A-5206 TS Hyload roller bearing supports the radial load at

[1] Neglecting the strengthening effect of the splines. For a more accurate computation, see R. L. Benford, Bending Stiffness of Beams with Ribs or Slots, *Machine Design*, February, 1947, 158.

the rear end of the shaft. This bearing is of ample capacity to carry this load at the rated speed.

A shaft seal either of mechanical type or packing may be provided. Provisions must be made to relieve leakage escaping into the bearing housings at both ends. This may be done by slippage taps on the housings or by drilling connections between the suction chamber of the pump and the bearing housings. In the latter case, care must be taken to have the shaft packing absolutely airtight.

The manufacture of the pump requires great accuracy and close tolerances. The vanes must fit in the slots within a few ten-thousandths of an inch. Not more than 0.001 to 0.002 in. total clearance should be allowed between rotor and end housings.

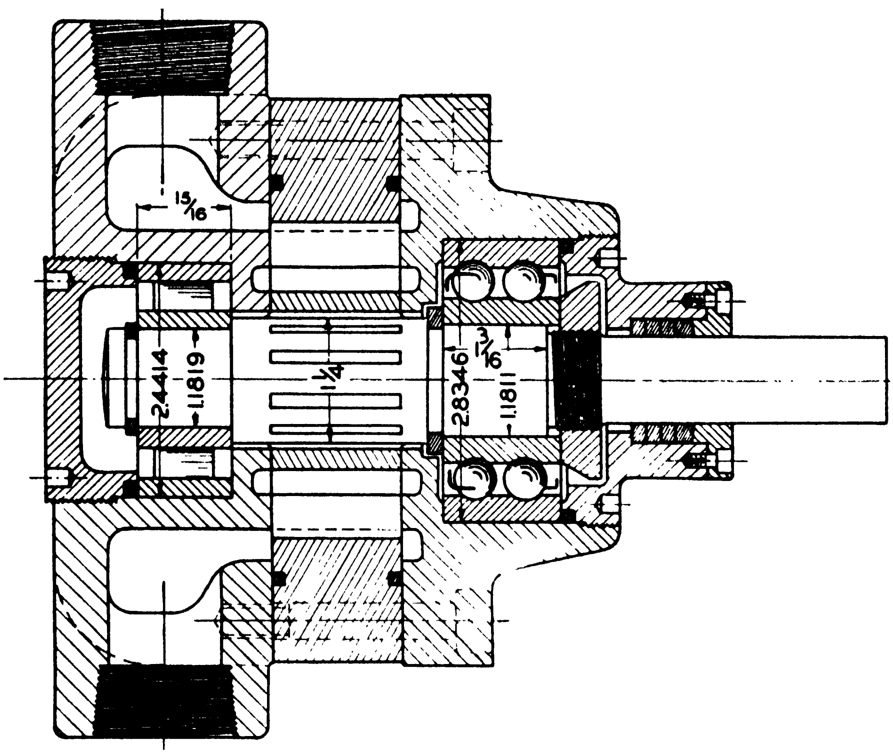


Fig. 64. Vane pump.

Illustration and Description of Commercially Available Vane Pumps

Vickers Vane Pump. The classical example of vane-pump design is the well-known Vickers pump. By skillful design, superior manufacturing methods, and excellent promotional work, Vickers Incorporated has become the outstanding leader in the field of applied hydraulics. Their vane pump is the nucleus about which the entire line of over 5,000 standardized units is built.

The pump, as illustrated in Fig. 65, features a fully balanced rotor design with two intakes and two outlets diametrically opposite each other. The side plates are brass castings with extensions fitting closely in pump body and end cover. These extensions serve as bearings for the

rotor. Being in full hydraulic balance, the rotor does not require anti-friction bearing mounting. The drive shaft runs in separate ball bearings and is splined in the rotor so that it may not transmit misalignment to the rotor. The pressure chamber is elliptoid with sections at the lap spaces forming true arcs so that there is no movement of the vanes when they are subjected to the hydraulic load.

The pumps are recommended for maximum pressure of 1,000 psi for standard models and 2,000 psi for high-pressure models consisting of two pumps in series. Recommended oil viscosity is from 150 to 225 SSU at

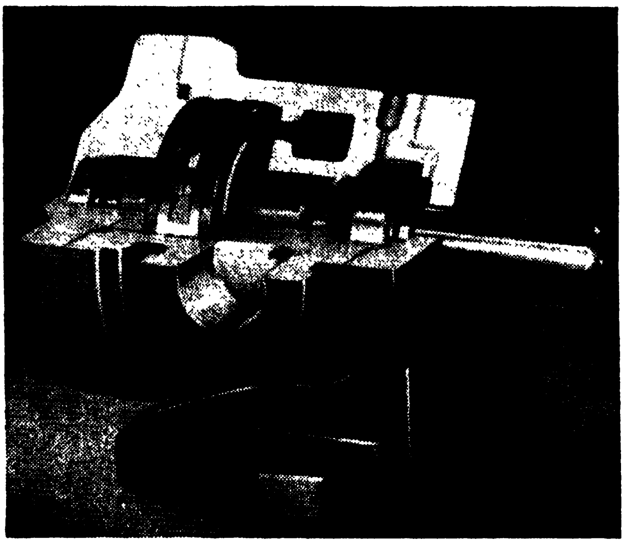


FIG. 65. Balanced vane pump. (*Vickers Inc., Detroit.*)

100°F. Maximum recommended oil temperature is 155°F. Speeds are from 1,800 to 1,200 rpm, according to size. Excellent efficiencies, both volumetric and mechanical, are obtained. Medium-size pumps show 85 per cent over-all efficiency through almost the entire working range. Volumetric efficiency of the very small pumps is lower.

A great variety of sizes and styles is available. There are three series: small, from 1 to 11 gpm; medium, from 14 to 36 gpm; and large, from 40 to 60 gpm. Intermediate capacities within each series fill any desired requirement. Pumps may be supplied with flanged or threaded pipe connections and with flange or foot mounting. Mounting dimensions of the medium-series pump with threaded connections are shown in Fig. 66, which conveys an impression of the small and compact size of this pump.

In addition to the single units a great number of combinations of pumps are available. These combinations consist of two pumping units mounted in one body casting. These units may be connected in a number of ways and provided with automatic controls to obtain different operating characteristics as follows:

1. Two units in parallel with either independent or combined discharge. Large units of combined discharge of as much as 120 gpm are available.

2. Two units in parallel with unequal capacity, by which arrangement the larger volume pump is unloaded at an intermediate pressure, and the small-volume pump carries on to the maximum pressure. Two series of combinations are available with numerous variations of pressure and

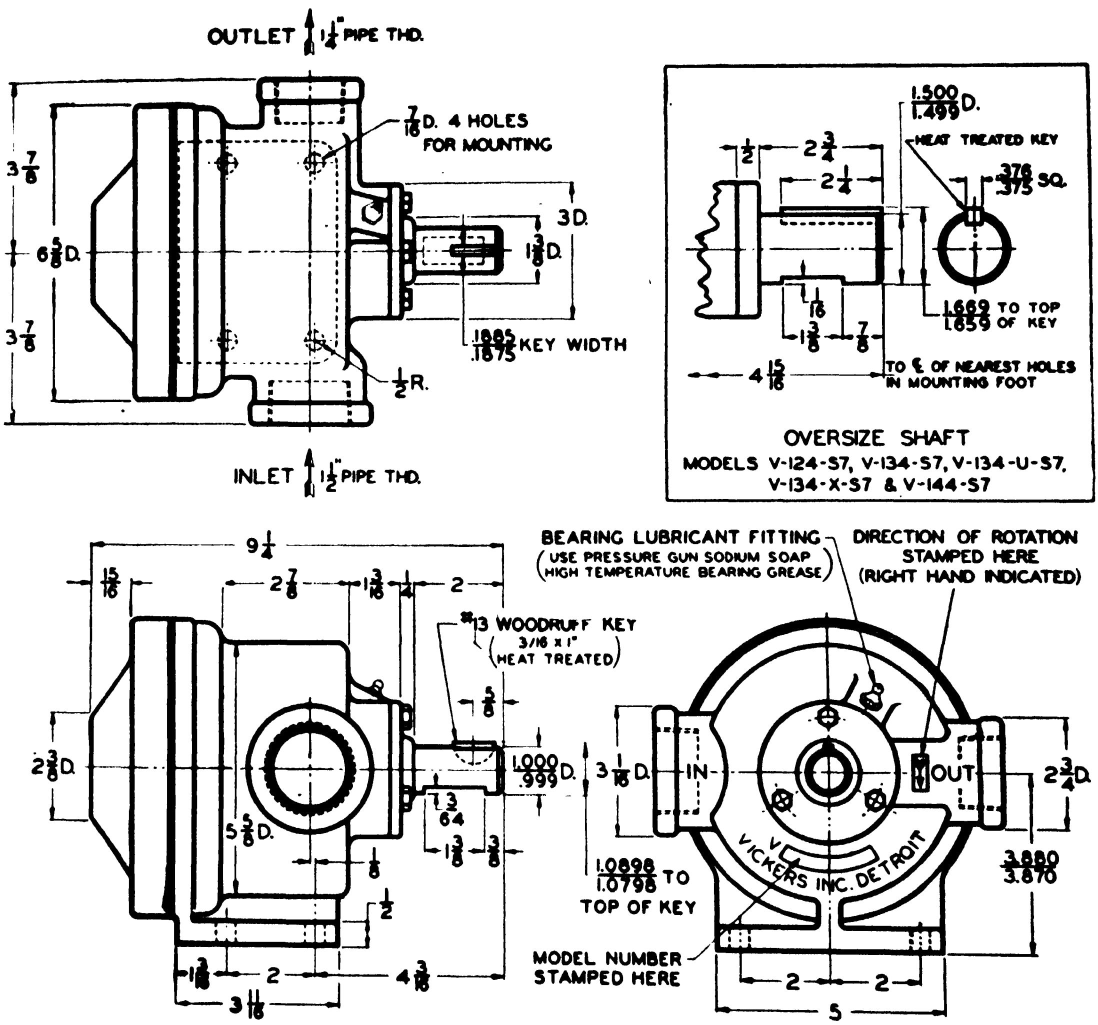


FIG. 66. Mounting dimensions of medium-series vane pumps. (*Vickers Inc.. Detroit.*)

delivery of both components. These pump combinations are especially suitable where it is desired to advance a hydraulic ram or piston rapidly at moderate pressure and follow at a slow-speed squeeze at high pressure.

3. Two units in series with substantially equal capacity to obtain double the operating pressure of a single unit. These combinations also are available in three series: small, from 2 to 7 gpm; intermediate, from 14 to 18 gpm; and large, from 37 to 57 gpm.

4. Series-parallel connected units. At pressures up to 1,000 psi, both units operate in parallel. At a preselected pressure of 1,000 psi or

less, the pumps are automatically shifted into series operation at one-half the output and will deliver up to 2,000 psi. Combined deliveries up to 120 gpm are available.

To effect automatically the delivery characteristics enumerated above, ingenious valves and controls have been developed, which will be fully described in Chap. X. These are combinations of Vickers Hydrocone relief valves, Hydrocushion unloading valves, sequence and load distribution, and check valves. Figure 67 shows a two-stage pump complete with integral unloading and relief valves.

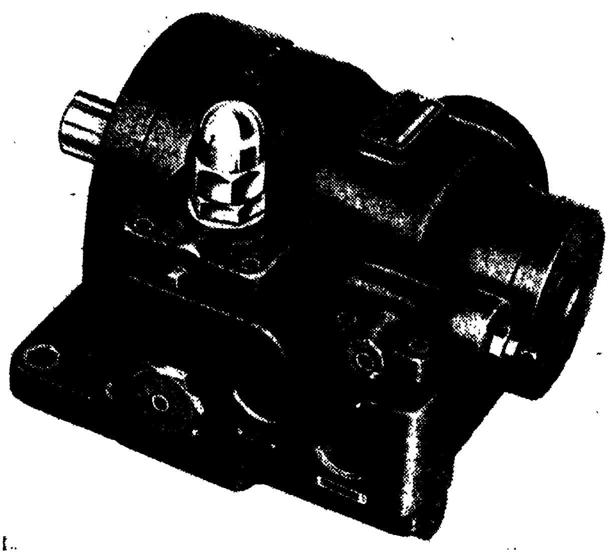


FIG. 67. Two-stage vane pump and valve combination. (*Vickers Inc., Detroit.*)

Racine Tool and Machine Co. A very unique type of variable-delivery vane pump is manufactured by the Racine Tool and Machine Co. The pump features a shiftable pressure chamber to vary the displacement. The pressure chamber is carried on thrust blocks operating on roller bearings to permit shifting at high pressure. The rotor is splined on the drive shaft. The pump is unbalanced, and the full hydraulic load must be carried by the shaft bearings, which are ample and sturdy. A section of the pump is shown in Fig. 68. Vane slots are slanted to permit use of longer, better supported vanes. The rotor is alloy steel, hardened and ground. Assembly of shaft, bearings, and rotor together with vanes is shown in Fig. 69.

The Racine pump uses bronze surface-ground port plates. Figure 70 shows the porting. The lower slots connect to the bottoms of the vanes, and ducts lead from the ports to the lower slots, whereby the vanes are balanced. The pump is available in three capacities: 12, 20, and 30 gpm at 1,200 rpm. Maximum recommended pressure is 1,000 psi. Oil of 200 SSU viscosity at 100°F is recommended.

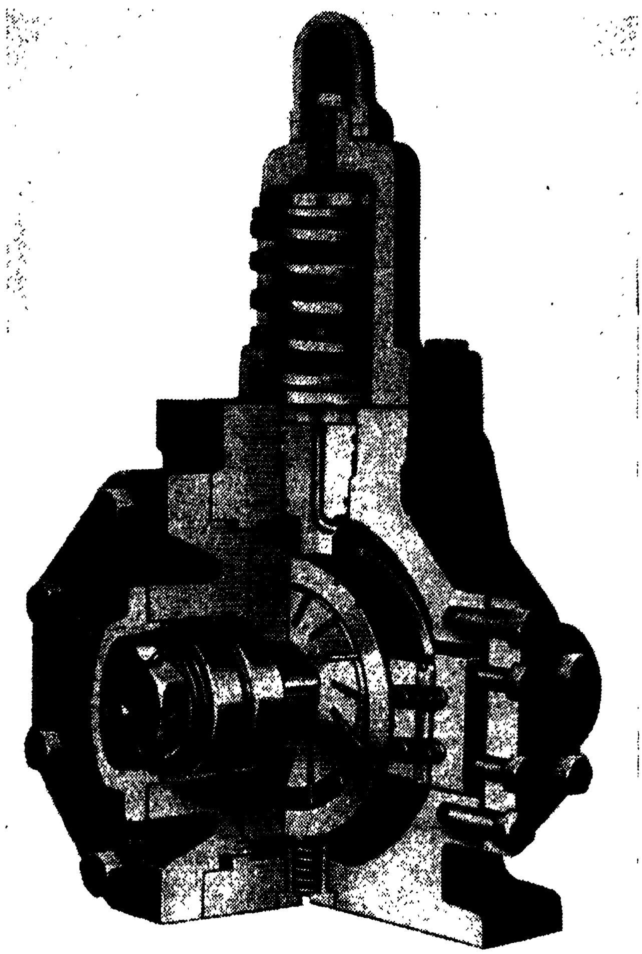


FIG. 68. Variable-delivery vane pump. (*Racine Tool and Machine Co., Racine, Wis.*)

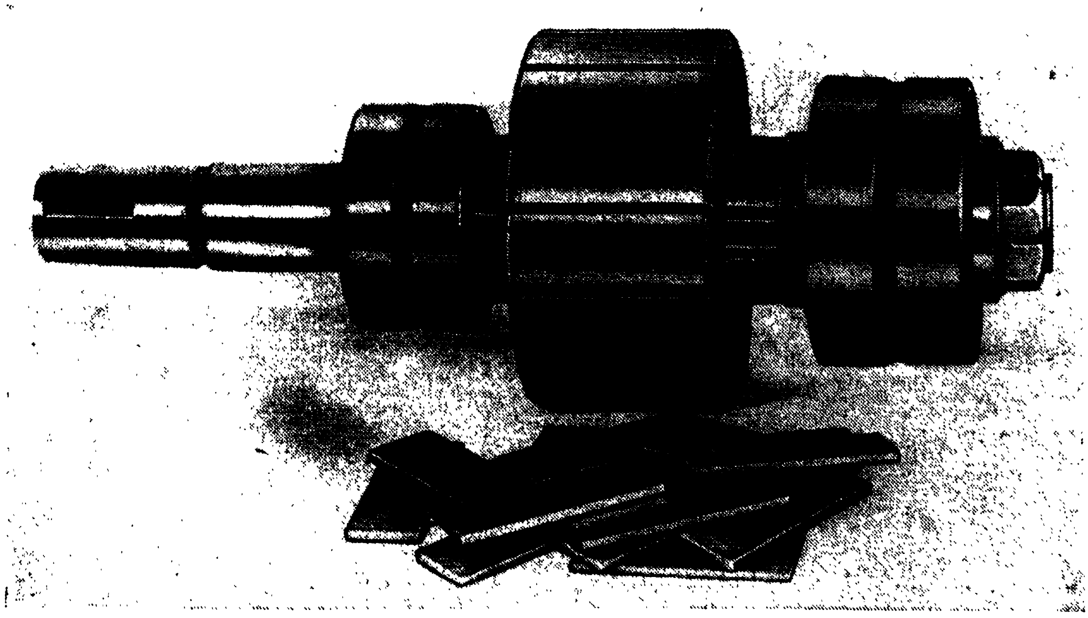


FIG. 69. Shaft and rotor assembly of variable-delivery vane pump. (*Racine Tool and Machine Co., Racine, Wis.*)

The pump may be equipped with an automatic volume control as shown in Fig. 68. A small spring shown at the bottom tends to force the pressure chamber toward the neutral or no-delivery position. This action is opposed by a heavy dual spring acting against a differential plunger. The stroke of the heavy spring is limited by engagement of the bottom washer on the housing face. Hydraulic pressure is piped from the discharge port of the pump to the differential area and opposes the action of the spring. When the hydraulic pressure in the pump has reached the value that corresponds to the setting of the spring, the differential plunger will move and permit the pressure chamber to assume an approximately neutral position, with just enough delivery of the pump to maintain leakage loss at the set pressure. Thus the pump is capable of maintaining a "deadhead"

FIG. 70. Port plate for variable-delivery vane pump. (*Racine Tool and Machine Co., Racine, Wis.*)

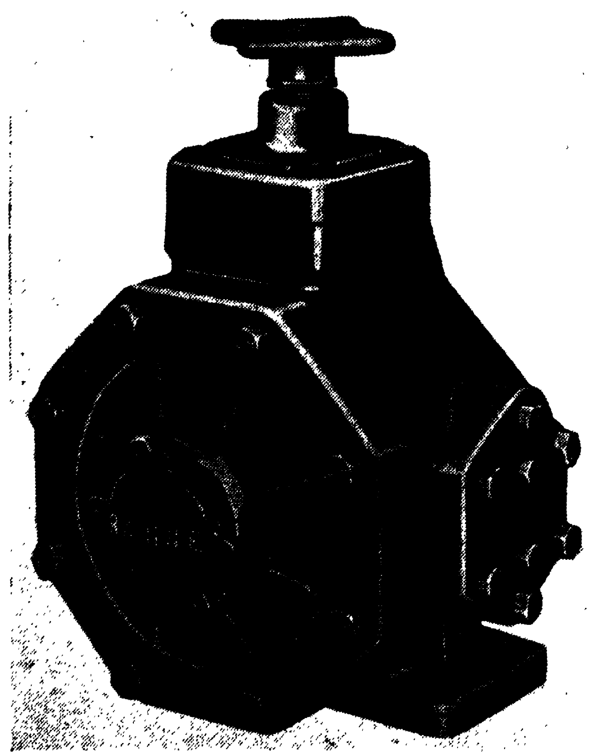


FIG. 71. Variable-delivery vane pump with adjustable volume control. (*Racine Tool and Machine Co., Racine, Wis.*)

of pressure at a minimum of discharge, greatly reducing power consumption and heating.

Other control options are available, such as an adjustable volume control for preset output of any desired capacity. An exterior view of the pump equipped with this control is shown in Fig. 71. Dual-pressure controls for two selective pressures, either by manual or by remote electric control, are also available.

4. RADIAL AND AXIAL PLUNGER PUMPS

Radial Plunger Pumps

Most radial plunger pumps now commercially available are of the variable-delivery type. This is due to the fact that radial-plunger-pump design lends itself readily to adoption of the variable-delivery principle without substantial increase in cost. The pumps are of the positive-displacement type and are capable of generating high pressures in the neighborhood of 2,500 to 3,000 psi. The variable-delivery principle has many advantages, such as change in pump output with constant motor speed and without loss due to throttling of oil and by-passing. Most of the pumps are reversible, so that flow of oil may be reversed without reversing valves in the hydraulic system and without reversing the driving motor. With suitable controls the pump is well adapted for the requirements of heavy-duty hydraulic service, such as with presses, steel-mill machinery, and the like. Accurate control of delivery has led to its adoption for machine-tool drives, broaching machines, etc. Combinations of pumps and hydraulic motors are used for transmissions, which will be discussed in Chap. VIII.

The operating principle of a radial, variable-delivery, reversible-discharge pump is shown in Fig. 72. The pump consists essentially of a cylinder rotor revolving about a central valve spindle. The cylinder rotor is revolved by means of a drive shaft connected to it. Oil is supplied through passages consisting of bored holes in the valve spindle terminating in a circumferential valve slot extending over an arc of 180° less the lap space. Similarly, discharge takes place through another set of passages opposite the suction port having a similar valve slot. Pistons are mounted in cylinder bores in the cylinder rotor and arranged to reciprocate, traveling outward while passing the suction slot to draw in pressing fluid and inward during the discharge stroke to discharge pressing fluid. On their outer ends, the pistons have suitable thrust shoes or other similar means, whereby their reaction is transmitted to an outer rotor or reaction ring that revolves in suitable bearings in a retaining ring or sliding block. The means used to transmit the reaction of the pistons to

the outer rotor is the most important distinguishing feature between the different makes of pumps, which are otherwise very similar in principle and operation.

If the outer reaction rotor is shifted so that its center coincides with the center of the cylinder rotor, no movement of the pistons will take place. This condition is shown under *B* in Fig. 72.

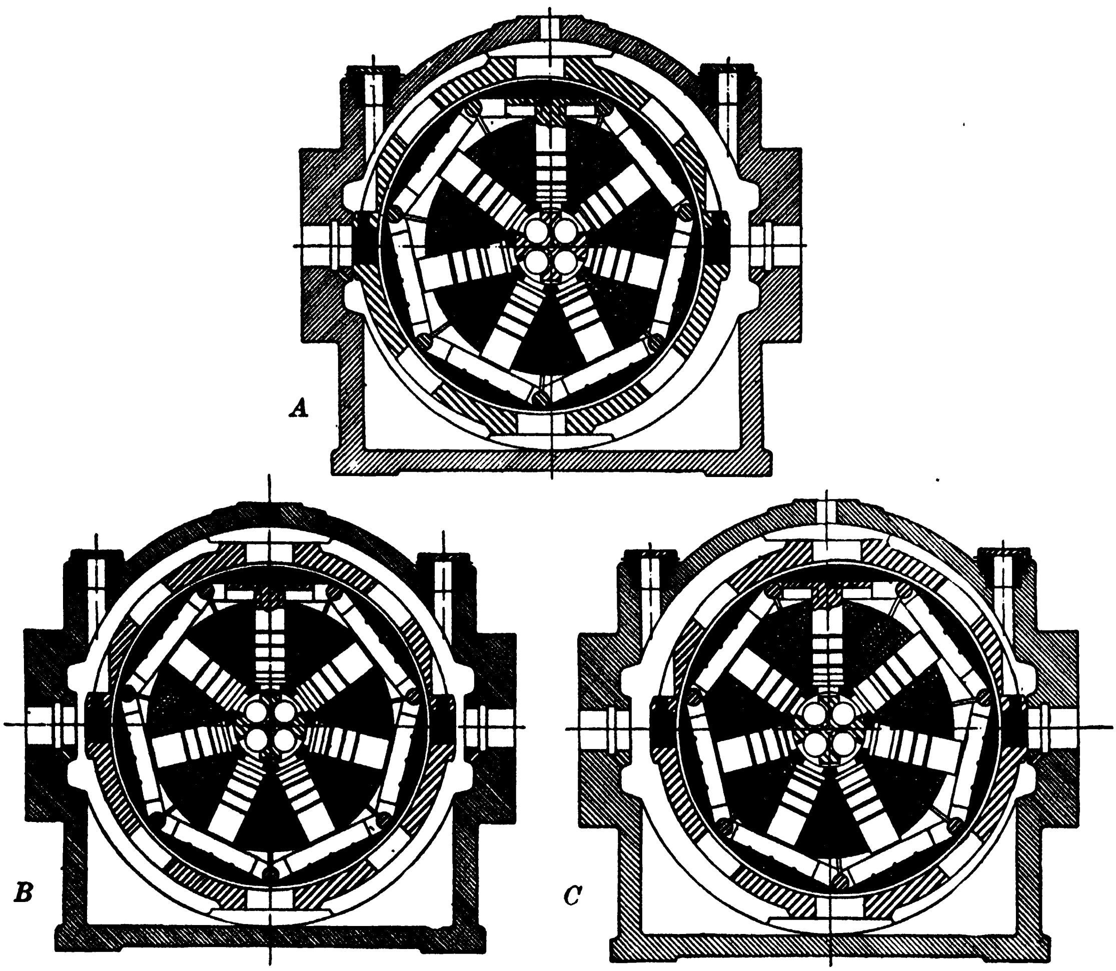


FIG. 72. Operating principle of variable-delivery pump. (*Hydro-Power Inc.*)

If the outer ring is shifted off center, as shown in *A* in Fig. 72, reciprocation of the plungers will take place. Beginning at dead center in a horizontal position, the plungers will travel outward over a 180° arc, taking in fluid from the corresponding passages; then they will travel inward over the remaining arc of 180°, expelling fluid through the discharge passages. The amount by which the outer ring is shifted off center determines the stroke of the pistons, which is double the eccentricity. By shifting the outer ring off center in the opposite direction, as shown at *C* (Fig. 72), suction and discharge will be reversed, provided that direction of rotation is maintained.

With the design of crosshead shown in Fig. 72, the pistons will have a pure harmonic motion, as will be shown in the following. Figure 73 shows a piston in one of its intermediate positions. The piston revolves with the cylinder rotor with a constant angular velocity ω. Eccentricity between cylinder and reaction rotor is denoted as e (in inches). Then we have

$$x = e - e \cos \omega t \tag{24}$$

where x is the stroke of the piston at the angle ωt.

The instantaneous piston speed, and with it the rate of flow, may be computed as follows:

$$\frac{dx}{dt} = \omega e \sin \omega t \qquad \text{in./sec} \tag{25}$$

With the area of one piston A in square inches, we have instantaneous output of one piston

$$Q_l = A\omega e \sin \omega t \qquad \text{cu in. per sec} \tag{26}$$

In a multiple-plunger pump, the total instantaneous output is the sum of the instantaneous capacities of the pistons, expressed in cubic inches per second. This figure may be plotted in a graph as function of the crank angle ωt.

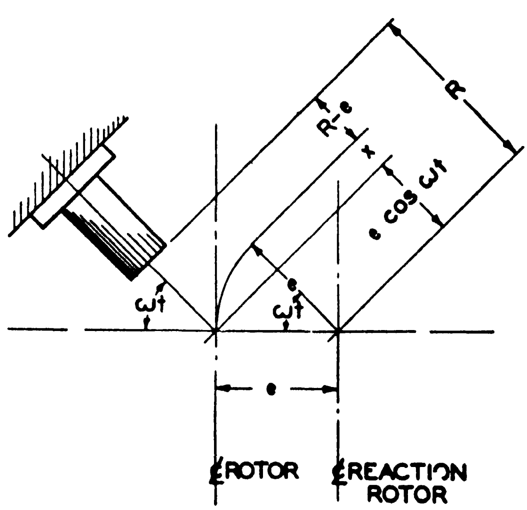


FIG. 73. Piston movement in radial pump.

If this is done, it will be found that surges in the rate of flow are more severe and less frequent in a pump with an even number of cylinders. Pumps having an odd number of cylinders give peaks in the rate of flow equal to twice the number of cylinders per 180°, while in the case of even numbers, there are as many peaks in 180° as there are cylinders. For instance, in a six-cylinder pump there are six peaks in 180° (counting both positive and negative variations from the average), while in a five-cylinder pump there are ten. The peaks in the six-cylinder pump occur at multiples of 30°, while the peaks in a five-cylinder pump occur at multiples of 18°.

The instantaneous output of all cylinders in a multiple-plunger pump is

$$Q_l = A\omega e \sum_{i=1}^{i=N} \sin\left(\omega t + \frac{2\pi i}{N}\right) \tag{27}$$

Where N denotes the total number of plungers. This is true only for positive values of $\sin[\omega t + (2\pi i/N)]$.

The mean output of a multiple-plunger pump is

$$Q_M = \frac{A\omega eN}{\pi} \quad \text{cu in per sec} \tag{28}$$

$$\text{Output variation} = \frac{Q_I}{Q_M} = \frac{\pi}{N}\sum_{i=1}^{i=N} \sin\left(\omega t + \frac{2\pi i}{N}\right) \tag{29}$$

For a six-plunger pump at 30°, for instance, $Q_I/Q_M = 1.045$ or a 4½ per cent variation in output. At 60°, $Q_I/Q_M = 0.906$, which is a 9.4 per cent variation from the mean. Contrasted with this, a five-plunger pump

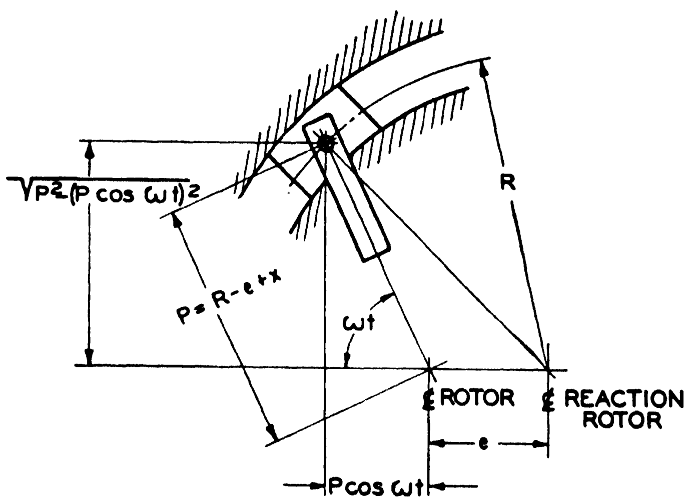


FIG. 74. Piston movement in radial pump.

shows at 18° $Q_I/Q_M = 1.015$ or 1½ per cent variation, and at 36° $Q_I/Q_M = 0.965$ or 3½ per cent variation from the mean.

This demonstrates the obvious advantage of an odd number of cylinders over an even number, and all commercial radial plunger pumps have an odd number of pistons.

We have shown that the type of crosshead construction illustrated in Fig. 72 gives a pure harmonic motion. This is not true for all designs of radial pumps. The Hele-Shaw variable-delivery pump employs a design of piston and crosshead in which the piston is carried on a gudgeon pin at its outer end, which in turn is mounted in two sliding segments in the reaction rotor. This is illustrated diagrammatically in Fig. 74.

We have

$$R^2 = P^2 - (P\cos\omega t)^2 + (e + P\cos\omega t)^2 \tag{30}$$

or

$$R^2 = P^2 + e^2 + 2eP\cos\omega t$$

From this,

$$P^2 + 2eP \cos \omega t + e^2 - R^2 = 0$$

and

$$P = -e \cos \omega t \pm \sqrt{e^2 \cos^2 \omega t - e^2 + R^2} \tag{31}$$

or

$$P = -e \cos \omega t \pm \sqrt{R^2 - e^2 \sin^2 \omega t} \tag{32}$$

To investigate the influence of the sine function ωt, we shall express the radical as an infinite converging series as follows:

$$P = -e \cos \omega t + R\sqrt{1 - \frac{e^2}{R^2} \sin^2 \omega t} \tag{33}$$

$$= -e \cos \omega t + R\left(1 - \frac{e^2}{2R^2} \sin^2 \omega t - \frac{e^4}{8R^4} \sin^4 \omega t \cdot \cdot \cdot\right) \tag{34}$$

Retaining only the first two members in the series, we obtain

$$P = R\left[1 - \frac{1}{2}\left(\frac{e}{R}\right)^2 \sin^2 \omega t\right] - e \cos \omega t \tag{35}$$

$$x = e + R\left[1 - \frac{1}{2}\left(\frac{e}{R}\right)^2 \sin^2 \omega t\right] - R - e \cos \omega t \tag{36}$$

for x increasing with ωt, that is, for suction stroke. For x decreasing with ωt (pressure stroke), x becomes

$$x = e - R\left[1 - \frac{1}{2}\left(\frac{e}{R}\right)^2 \sin^2 \omega t\right] + R - e \cos \omega t \tag{37}$$

For $\frac{1}{2}\,(e/R)^2 \sin^2 \omega t = 0$, Eqs. (36) and (37) revert to Eq. (24), that is, in case of an infinitely long connecting rod. With $R = 10e$ the correction amounts to 0.5 per cent for the maximum value of $\sin^2 \omega t$.

The instantaneous output of one piston may be computed again as follows:

$$Q_I = A\frac{dx}{dt} = A\left(\omega \frac{e^2}{R} \sin \omega t \cos \omega t + \omega e \sin \omega t\right)$$

$$Q_I = A\omega e\left(\frac{e}{R} \sin \omega t \cos \omega t + \sin \omega t\right) \tag{38}$$

The output variation follows from this again:

$$\frac{Q_I}{Q_M} = \frac{\pi}{N}\sum_{i=1}^{i=N} \frac{e}{R} \sin\left(\omega t + \frac{2\pi i}{N}\right) \cos\left(\omega t + \frac{2\pi i}{N}\right) + \sin\left(\omega t + \frac{2\pi i}{N}\right) \tag{39}$$

Equations (26) and (38) may be employed to analyze the instantaneous output of any of the commercially available pumps operating either with a pure harmonic motion or the equivalent of a connecting rod of finite length.

Speeds and Capacities. Operating speeds of radial pumps range from 1,800 to 600 rpm, according to size. Pumps from 1- to 15-gpm capacity may be operated at from 1,800 to 1,200 rpm. Capacity ranges from 15 to 30 gpm require speeds of 1,200 to 900 rpm; 30 to 100 gpm, 900 to 720 rpm; 100 to 200 gpm, 720 to 600 rpm; over 200 gpm, 600 rpm or less.

Piston Diameters and Strokes. Practical experience on pumps actually designed indicates that good proportions are obtained with piston-stroke lengths from 60 to 65 per cent of piston diameters. Ordinarily from five to nine pistons are employed in pumps of this kind, although a much greater number is used in the Oilgear pump. The combinations shown in Table III should prove satisfactory.

TABLE III

Diam.	No. of pistons	Stroke ($2e$)
½	5	5/16
¾	5	½
1	5	⅝
1¼	7	⅞
1½	7	1
2	7	1¼
2½	7 or 9	1⅝

Capacity of pumps may be computed from Eq. (28), or, introducing n rpm and expressing Q in gallons per minute, we have

$$Q = \frac{2AneN}{231} \qquad \text{gpm} \tag{40}$$

Volumetric and mechanical efficiencies may be derived as shown in Sec. 2. Again we have

$$e_v = \frac{Q_0}{Q_g} \times 100 \qquad \text{in per cent} \tag{2}$$

$$e_m = \frac{HP_h}{HP_a} \times 100 \qquad \text{in per cent} \tag{3}$$

$$HP_h = 0.000583Q_g p$$

$$e_t = \frac{Q_0 \times 0.000583p}{HP_a} \tag{7}$$

Efficiencies of these pumps vary greatly with pressures, capacities, type of design and make of pumps, and oils used. Volumetric efficiencies range from 85 to 95 per cent at maximum rated pressure, increasing with decreasing pressure. Mechanical efficiencies in the normal rating range vary from 90 to 95 per cent.

Oils used in these pumps are generally of a heavier body than those employed in the gear or vane type of rotary pumps. Radial plunger pumps are often subjected to severe service and depend upon the maintenance of oil films for continued performance. Oils from 500 to 1,000 SSU at 100°F are recommended by most manufacturers for their product, with occasional recommendations as low as 150 and as high as 1,500 SSU for unusual conditions of pressure or temperature at either extreme of the scale.

Materials and Design. A radial pump is a precision piece of machinery and must be designed and manufactured as such. Tolerances and clearances given in Table I, Chap. V, are recommended for fits of pistons and pintle or valve spindle. For antifriction bearing mountings, the bearing manufacturer's recommendation should be followed. All other moving parts must be closely fitted with minimum running clearances and carefully machined for proper line-up so that the mechanism may function without binding and excessive friction.

The central valve spindle is a chromium-nickel forging, carburized and hardened. The portholes should allow for an oil velocity of from 5 to 10 ft per sec depending on size. Pintles may be fitted in the pintle cover by press fit or sliding fit, with nut for adjustment as in the Hydro-Power pump. Pintle end covers may be made from cast steel or alloy iron. Rotors are made from brass or bronze, manganese bronze being used where antifriction bearings are interposed between pintle and rotor and where bearing qualities are less important than hardness and tensile strength. Nickel-steel forgings are used for the pistons. These are also carburized and hardened.

An 80-10-10 bearing bronze serves well as material for piston-cross-head shoes or slippers in Hele-Shaw-type pumps. Reaction rotors may be made from steel castings or forgings, and one make of pump employs alloy iron for this purpose. A good grade of cast iron will suffice for housings and drive-shaft end covers. Revolving parts are mounted on antifriction bearings of the ball or roller type.

Example of Design Calculation: In the following a calculation will be carried out for a pump of 20 gpm capacity. Maximum discharge pressure is 2,500 psi. Assuming a total efficiency of 85 per cent, we have

$$HP_a = \frac{Q_0 \times 0.000583p}{e_t} = \frac{20 \times 0.000583 \times 2{,}500}{0.85} = 34\tfrac{1}{2}\text{ hp}$$

The geometric capacity of the pump at 92½ per cent volumetric efficiency is 20/0.925 = 21½ gpm. With Q_g = 21½ gpm, n = 1,200 rpm, and N = 5, we have from Eq. (40)

$$Ae = \frac{21.5 \times 231}{2 \times 5 \times 1{,}200} = 0.415$$

With e = ⅜, A = 0.415/0.375 = 1.1, and piston diameter = 1$\frac{3}{16}$

$$\text{Suction pipe area at 5 ft per sec} = \frac{21.5 \times 231}{60 \times 5 \times 12} = 1.38 \text{ sq in.}$$

Since the pump is reversible and either one of the lines may be subjected to the full working pressure, Schedule 160 pipe should be used. One-and-a half-inch pipe will be required to provide the suction area.

Pintle ports are to be calculated for a velocity of 8 ft per sec.

$$\text{Diameter of holes} = \sqrt{\frac{4 \times 21.5 \times 231}{\pi \times 480 \times 12 \times 2}} = \tfrac{3}{4} \text{ in.}$$

With the four holes properly spaced, we obtain a pintle cross section as shown in Fig. 75. The pintle diameter becomes 2⅝. Figure 75 shows the five pistons spaced around the circumference of the pintle. At the bottom of the cylinder bores the diameter is reduced about 30 per cent, forming the intake holes from the valve slots into the cylinders. The valve slots are approximately the same width as the reduced diameter of the holes and are milled on opposite sides of the pintle with a lap space between them as indicated. Lap may be positive, zero, or negative, depending on whether it is wider than, equal to, or narrower than the reduced diameter of the cylinder bore. In this case, for instance, a lap ⅞ in. wide would be zero lap. Zero and negative laps result in a quieter running pump, owing to elimination of cavitation, but have slightly more slippage loss. Positive laps tend to make the pumps noisy, but are frequently used. One-inch wide lap is suggested as the limit for a positive lap on this pump.

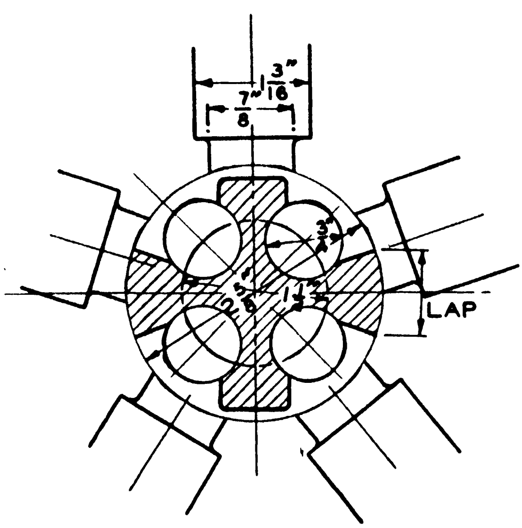


FIG. 75. Section of pintle and cylinder rotor.

The rotor revolves about the pintle, which serves as distributing valve and may be mounted in a different number of ways. Direct mounting upon the pintle without intervening bearings is used in one make of pump, described later, while others employ antifriction bearing mounting of the rotor, either on the pintle or in the end housings.

The pintle is subjected to an unbalanced load, the pressure half being subjected to the full hydraulic pressure across the port width and to a pressure gradient extending axially on both sides of the port. This load must be carried by the oil film under the opposite end or suction side of the pintle, when the rotor is directly mounted on the pintle. Where antifriction bearing mounting is resorted to, these bearings are called upon to support this load. In that case, the total amount of pressure may be limited

by balancing grooves cut into the rotor circumferentially, permitting the pressure to equalize after reaching the grooves. The area of unbalanced pressure is therefore bounded by the relief grooves in axial direction. The unbalanced load on the pintle may be approximated by assuming that one-half the total pressure is applied on an area equaling the diameter of the pintle times the distance between the balancing grooves. In this case, for instance, the balancing grooves are about 2 in. apart. The unbalanced load, therefore, is approximately

$$\frac{2{,}500 \times 2 \times 2\frac{5}{8}}{2} = 6{,}500 \text{ lb}$$

or 3,250 lb for each bearing. Bearing selection is based upon this load and speed.

Clearance between rotor and pintle should be from 0.001 to 0.0023 in. (see Table I, Chap. V). With these close clearances, obviously run-out on bearings must be held to a minimum, and precision-type bearings should be selected. Allowing for a suitable wall thickness of cylinder rotor, we determine the inside diameter of the reaction rotor by adding the stroke length plus clearance, to allow for shifting the reaction rotor an amount $= e = \frac{3}{8}$ in. Antifriction bearings are used for mounting reaction rotors on all makes of pumps. Referring to Fig. 75, these bearings may be subjected to the load of three pistons as follows:

$$L = Ap(1 + 2 \sin 18°) = Ap \times 1.618 = 1.1 \times 2{,}500 \times 1.618 = 4{,}450 \text{ lb}$$

The pistons are fitted with a clearance of 0.0006 to 0.0015 in. They should be guided in the cylinder rotor with a guide length of at least twice their diameter in neutral position of pump. The means by which the thrust of the pistons is transmitted to the reaction rotor vary in the different designs of pumps. Common to all of them is the necessity of providing for relative movement between piston crosshead and reaction rotor, so that the pistons may adjust themselves to the change in peripheral velocity as they revolve about the cylinder-rotor center at varying distances from the center.

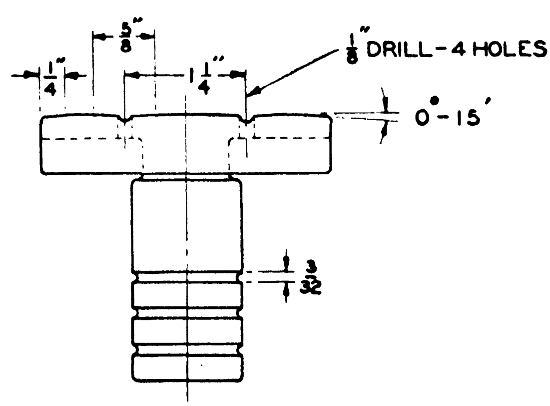


FIG. 76. Piston design for radial pump.

T-shaped pistons may be employed, bearing against flat reaction ways, as shown in Fig. 73, or pistons may be fitted with gudgeon pins mounted in segmental slippers guided in peripheral grooves in the reaction-rotor halves. Bearing pressure on cross-heads is about 500 psi. To permit establishment of a load-carrying oil film on these surfaces, they may be divided into a series of tapered and straight land spaces, as shown in Fig. 76. The tapered surfaces cause formation of a wedge-shaped oil film, which will satisfactorily support a load of 500 psi. Pistons are provided with balancing grooves, as shown in Fig. 76.

In order to permit the piston crossheads to operate in a bath of oil, the space around the crossheads must be made oiltight. This may be done by clamping the rotor halves together so as to form a casing, as shown in Fig. 77, or by providing a shell or casing between them. Slippage oil from the pistons will accumulate in this space and provide lubrication. The shift ring for the reaction rotor is mounted in suitable ways, whereby it may be shifted off center to produce eccentricity and pump stroke. Lubri-

cation of bearings and other operating parts is provided by the slippage oil escaping from the pistons and pintle. Ordinarily this amount of lubrication will suffice, but for extreme cases and heavy service, auxiliary lubrication is recommended. Slippage and lubricating oil are permitted to drain out of an opening in the bottom of the pump into a slippage tank, whence it is removed by the slippage or scavenging pump and returned to the oil-supply tank. Where pumps are mounted on oil-supply tanks, the slippage may drain directly into these tanks. Slippage pumps should be of sufficient capacity to take care of 15 to 25 per cent of the radial-pump capacity to prevent flooding after wear has taken place in the radial pump.

Description and Specifications of Commercially Available Units. *The Hele-Shaw Pump.* This pump, manufactured by the American Engineer-

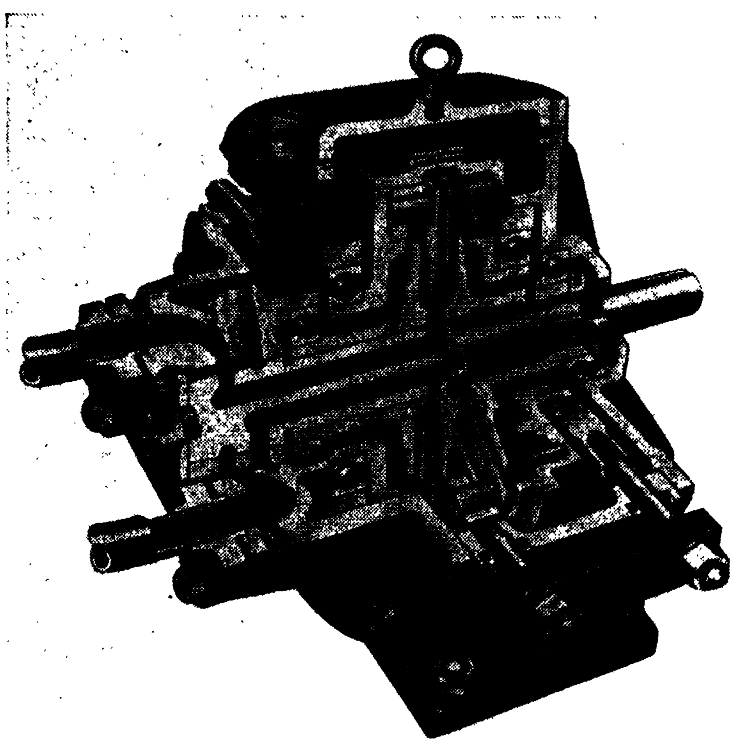


FIG. 77. The Hele-Shaw variable-delivery pump. (*American Engineering Co., Philadelphia, Pa.*)

ing Co., Philadelphia, Pa., is perhaps the oldest commercially used unit on the market. Designed by Dr. H. S. Hele-Shaw in England in the early years of the First World War for hydraulic operation of ships' steering gear and other auxiliary, the pump has been in successful production with few alterations or modifications in the basic design. This speaks well for the excellence of the original design and the genius of its inventor.

The kinematics of the pump correspond to those discussed in connection with Fig. 74. Pistons have wrist or gudgeon pins operating in segmental shoes called "slippers," which are mounted in the reaction-rotor halves. As shown in Fig. 77, the reaction rotor operates on ball

bearings and is caused to rotate by the friction of the slippers as the primary or cylinder rotor is turned by the drive shaft. The slippers care for the difference in peripheral speed, as the pistons operate at varying distances from the center of the cylinder rotor. It should be mentioned here that all radial pumps are in perfect dynamic balance, as the center of gravity of their rotating parts always coincides with their center of rotation, notwithstanding the fact that an eccentricity is produced between the two rotors. The reaction-rotor halves are semisteel castings, carefully machined and bolted together to form an oiltight casing for the operation of the slippers. Sliding blocks in which the reaction-rotor bearings are mounted serve to shift the reaction rotor. The sliding blocks operate in ways provided in the pump and covers. Rods threaded in the sliding blocks permit attachment of suitable devices on the outside of the casing to shift the blocks and reaction rotor in either direction to produce corresponding pumping action. The rotor is mounted in ball bearings in the pump end covers and rotates about the stationary pintle or valve spindle, which is supported in the pintle end cover in the manner of a cantilever beam. A pressure oil film is depended upon to support the closely fitted pintle inside the rotor.

The manufacturers recommend oils from 1,000 to 1,500 SSU viscosity for operation, according to severity of service. Oil temperatures must not exceed 110°F. Pressures as high as 3,000 psi may be obtained.

The pump is made in 10 high-pressure sizes ranging from 3¾ to 120 gpm at speed ranging from 1,200 rpm for the smallest to 600 rpm for the largest pumps. Pressures are from 3,000 to 2,000 psi. A low-pressure series is also available in nine sizes for a pressure of 1,200 psi and capacity up to 150 gpm. As in all radial pumps, suitable controls may be supplied to permit operation by manual or automatic control. They will be dealt with in Sec. 6 of this chapter.

The Hydro-Power Radial Pump. This pump, designed by the author, is manufactured by Hydro-Power Inc., Mount Gilead, Ohio. The pump, shown in Fig. 79, has a pure harmonic piston movement as analyzed in Fig. 73. The pistons are one-piece, T-shaped, alloy-steel forgings with the crossheads fitted in T-shaped thrust shoes, which are clamped between the reaction-rotor halves, as shown in Fig. 78.

The cylinder rotor is mounted directly on the central valve spindle or pintle by means of ultraprecision Timken roller bearings, preloaded so that there is no looseness or clearance within the bearing. Clearances and pintle deflection are so calculated that there is no contact between rotor and pintle, even at heavy loading, and consequently wear or seizing between these parts is impossible. The original setup of the pintle bearings may be restored or maintained by means of the adjusting nut on the

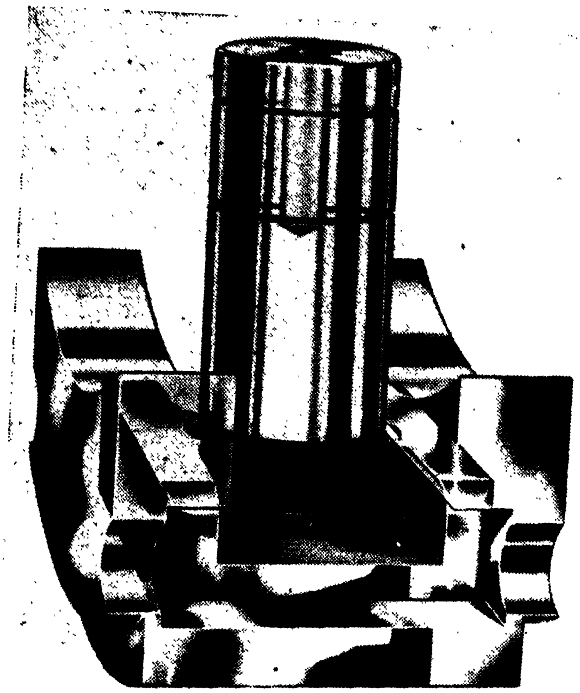


FIG. 78. Piston and guide block. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

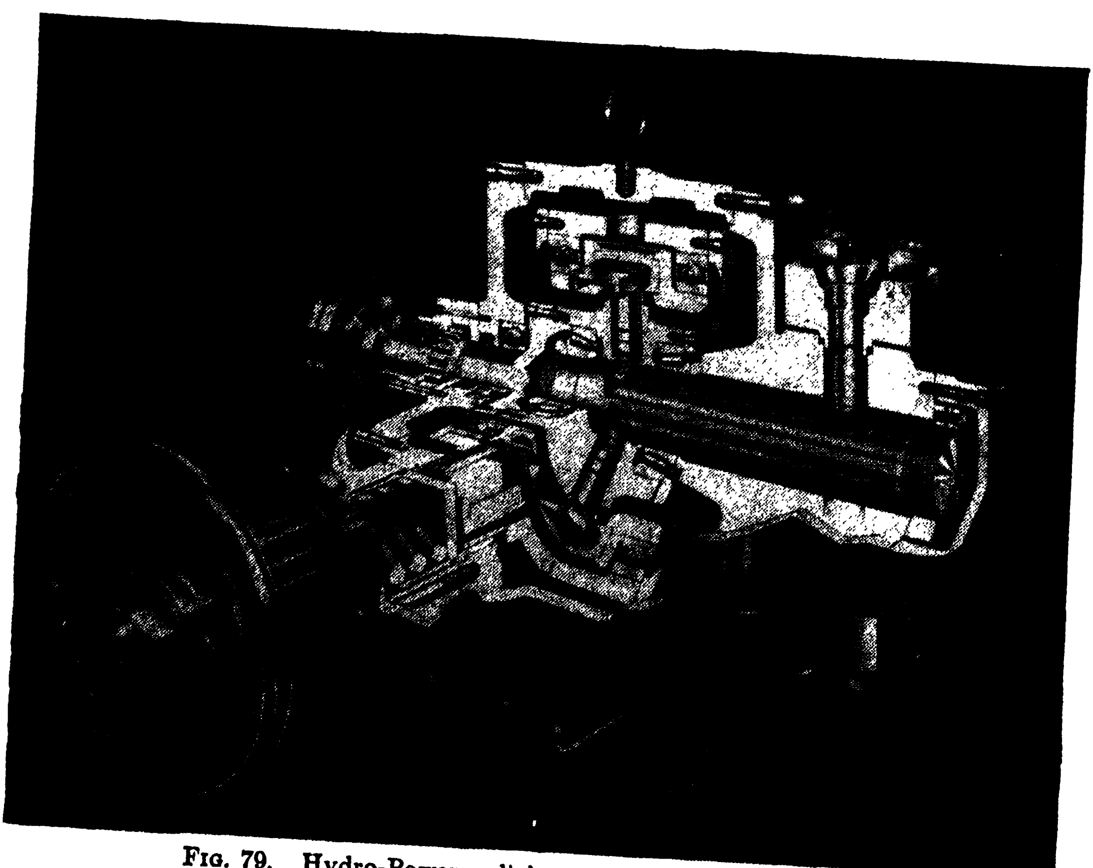


FIG. 79. Hydro-Power radial pump. (*Hydro-Power Inc.*)

outside of the pintle hub, by which the pintle may be drawn toward the pintle end cover, thus taking up any looseness in the bearings. The pintle is a close sliding fit in the pintle end cover and has passages and valve slots of conventional design. The shift ring, which contains the reaction-rotor assembly, may be shifted, as shown in Fig. 79, by means of control rods threaded in suitable bosses and extending to the outside of the casing.

Various controls may be applied to operate the shift ring by manual or automatic means. The pump may be equipped with an integrally mounted slippage pump, as shown in Fig. 79. An ingenious arrangement permits driving an auxiliary pump from one of the slippage-pump gears. The auxiliary pump provides hydraulic pressure for auxiliary lubrication, servo controls,

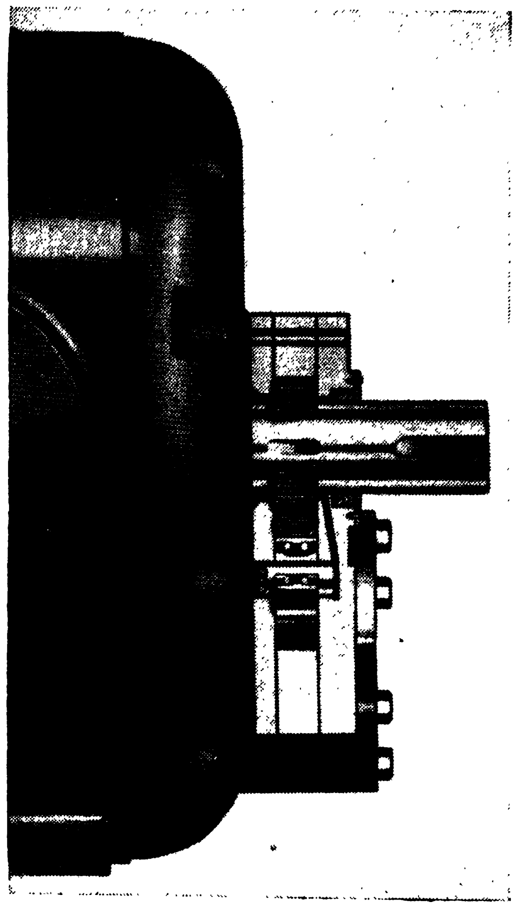


Fig. 80. Integral slippage pump. (*Hydro-Power Inc.*)

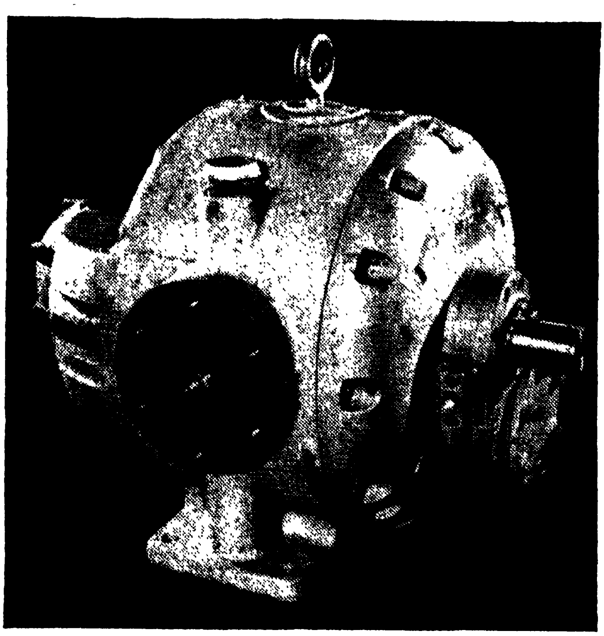
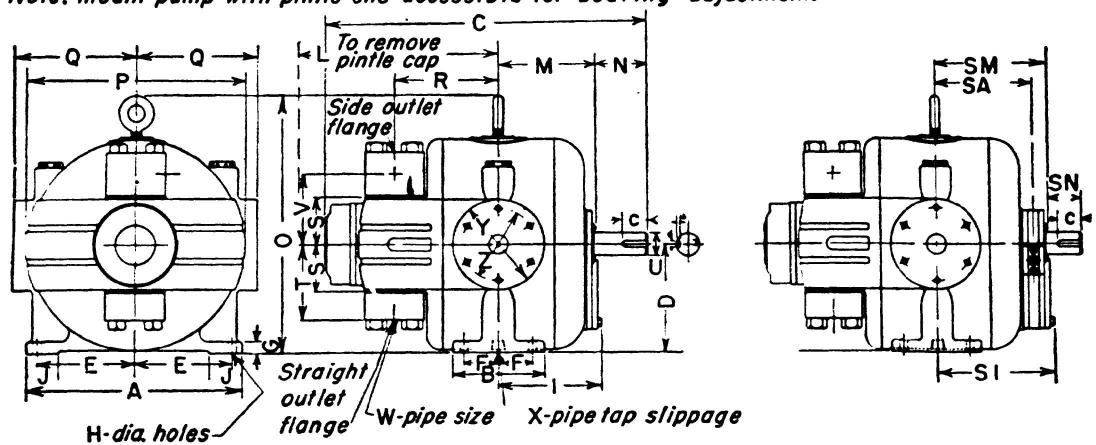


Fig. 81. The Hydro-Power radial pump. (*Hydro-Power Inc.*)

TABLE IV

Model and size	Rpm	Gpm	U	Keyway a	Keyway b	Keyway c	A	B	C	D	E	F	G	H	I	J	I	M	N	O	P	Q	R	S	T	V	W Suction	W Discharge
R-2	1,200	1¼	¾	3⁄16	3⁄32	17⁄16	6¾	3¼	12¼	3⅝	2⅞	1¼	½	7⁄16	3⅝	1⅜	79⁄16	315⁄16	17⁄16	10	7¾	315⁄16	3¾	1¾	....	.....	½ Tap	½ Tap
R-5 & 10	1,200	3¼ & 5	1	¼	⅛	1⅜	9½	3¼	16½	4⅞	4⅛	1	¾	9⁄16	4¾	2	9¾	5⅛	2⅜	12⅞	10	5¼	5⅜	2¼	3⅜	3¼	¾	½
R-20	1,200	11	1¼	⅜	3⁄16	1¾	11½	6	20¾	7	5	2¼	¾	11⁄16	6⅝	2¾	12⅜	6⅝	2⅞	18⅛	13¾	75⁄16	7	2⅝	4	3¾	1	¾
R-50	900	23	1¾	½	¼	2	16	7	25¾	9	7	2	1½	15⁄16	73⁄16	2¾	15½	91⁄16	2½	23½	17⅞	913⁄16	813⁄16	4	6	5½	1½	1¼
R-100	900	50	2¼	⅝	5⁄16	2½	20	8½	329⁄16	11¼	8¾	3⅛	1½	11⁄16	....	4½	20	.....	....	28	22⅛	121⁄16	115⁄16	5¼	7⅞	75⁄16	2½	2
R-200	900	95	3	¾	⅜	3	27	11½	4011⁄16	13¾	12¼	4¼	1½	11⁄16	....	4	24½	.....	....	33½	26¾	155⁄16	133⁄16	6	9⅝	8⅜	3	2½
R-400	720	185	4	1	½	4⅛	36	14	533⁄16	18¾	16½	5½	2	13⁄16	....	5	29⅜	.....	....	43	37	19	1613⁄16	7¼	11⅜	10½	4	3½

Note: mount pump with pintle end accessible for bearing adjustment.



Note: straight or side outlet flanges to be furnished as desired.

FIG. 82. Dimensions of Hydro-Power radial pumps. (*Hydro-Power Inc.*)

and other purposes. An exterior view of the pump is shown in Fig. 81.

The pumps are being manufactured in a range of sizes from 1¼ to 185 gpm. One-way and reversible models are available. The pumps are conservatively rated at 2,500 psi, but have successfully operated at 3,000 psi and higher. The full range of variable-delivery pumps is shown in Fig. 82, containing the essential dimensions and pertinent data.

For heavy-duty service, oil of 750 to 1,000 SSU at 100°F is recommended. Ordinarily, maximum oil temperature recommended is 120°F. Pumps have been successfully operated at temperatures as high as 150°F.

The Oilgear Pump. This pump is manufactured by the Oilgear Company, Milwaukee, Wis. The company is a pioneer in the introduction of fluid power feeds and their industrial applications. The pump has undergone a process of evolution and simplification culminating in the present design, based on the rolling-piston principle. A single, simple piece of ingeniously machined steel in the form of a rolling piston reacting on a steel ring has reduced the number of working parts to a minimum and makes possible smooth quiet and positive reciprocating motion at high speeds and varying strokes. Both variable- and fixed-displacement units are available.

VARIABLE-DISPLACEMENT UNITS. The variable-displacement units, as shown in Fig. 83, consist of a pintle, a cylinder barrel with seven or more closely fitted pistons, one or more reaction rings, and a rotor and slide block. The cylinder barrel, lined with an antifriction metal, rotates on a fixed, closely fitted pintle, made of alloy steel, hardened and ground, which is pressed into the case. A floating coupling shaft, splined to the input shaft, which is mounted on antifriction bearings, drives the cylinder barrel. Centrifugal force, which may be augmented by back pressure from the built-in gear pump or from return oil, keeps the convex surfaces of the rolling pistons (made of roller-bearing alloy steel, hardened and ground) against the concave surface in the reaction ring at all times. The rotor and rotor head, which contain the reaction rings, are mounted on large antifriction bearings and rotate with the cylinder barrel through contact of the rolling pistons against the reaction rings. A slide block mounted between four horizontal ways in the case and connected to the control mechanism carries the complete rotor unit and is used to vary the stroke of the pistons.

In Fig. 83 a handwheel screw control is shown that consists of a large screw flanged to the slide block, a long working nut, a handwheel, and lock nut. This handwheel provides accurate control of slide-block position and hence of the volume of oil displaced. Compression springs opposing the control hold the slide block firmly against the control

mechanism. Other controls are available for either manual or automatic actuation of the shift block; these will be discussed in Sec. 6 of this chapter.

A gear pump is built in the front housing for operating hydraulic controls, lubricating cylinder and pintle, and supplying make-up oil to the main system. On one-way variable-displacement pumps, a combined suction and return valve is flanged to the bottom of the case. Two-way

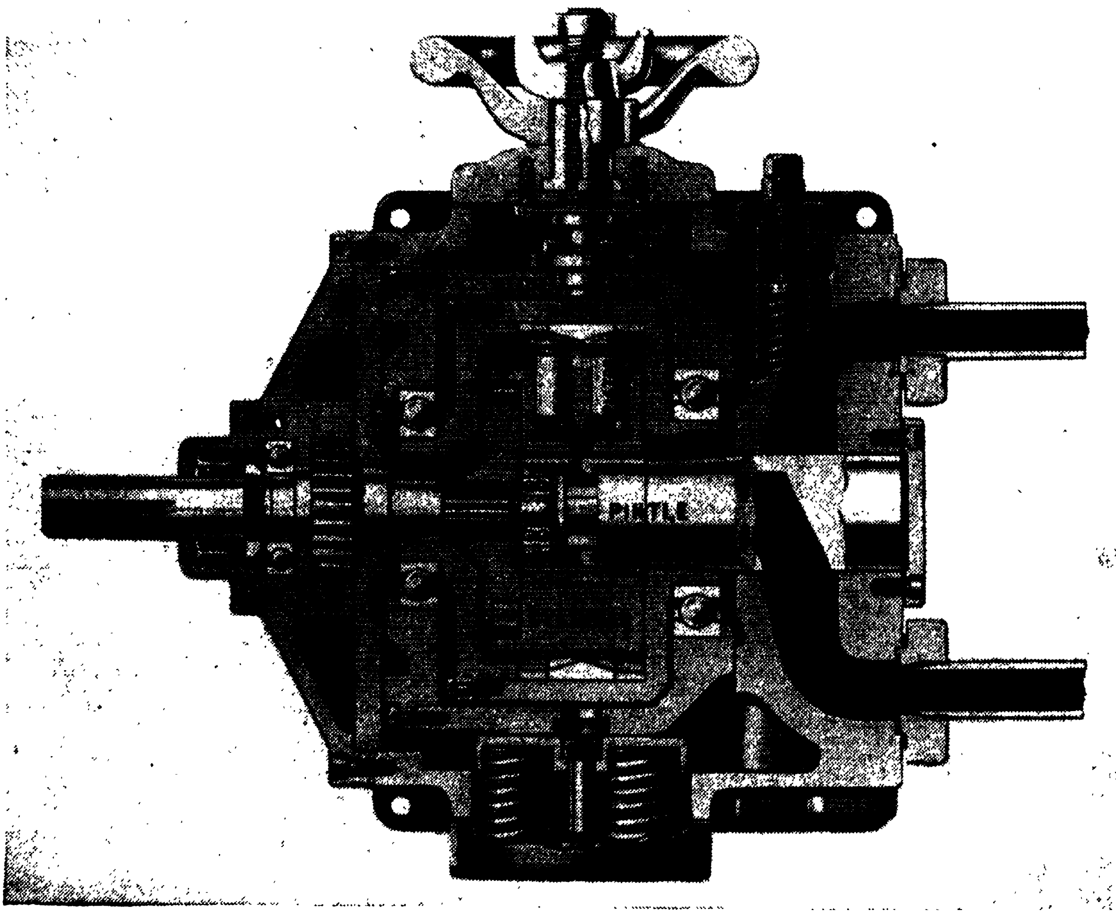


FIG. 83. Oilgear variable-delivery pump. (*The Oilgear Co., Milwaukee, Wis.*)

variable-displacement pumps use an automatic two-way suction and return valve.

Pumps are equipped with built-in relief valves for limiting the pressure of the variable-displacement unit and auxiliary gear pump.

The operating principle is very similar to that of the other makes discussed in this chapter. A unique part is played by the rolling pistons and illustrated in the following. When the center lines of cylinder and rotor coincide (see Fig. 84), no reciprocating motion is imparted to pistons as unit rotates, so no oil is delivered. As the slide-block-and-rotor unit is moved to the left by control mechanism (see Fig. 85),

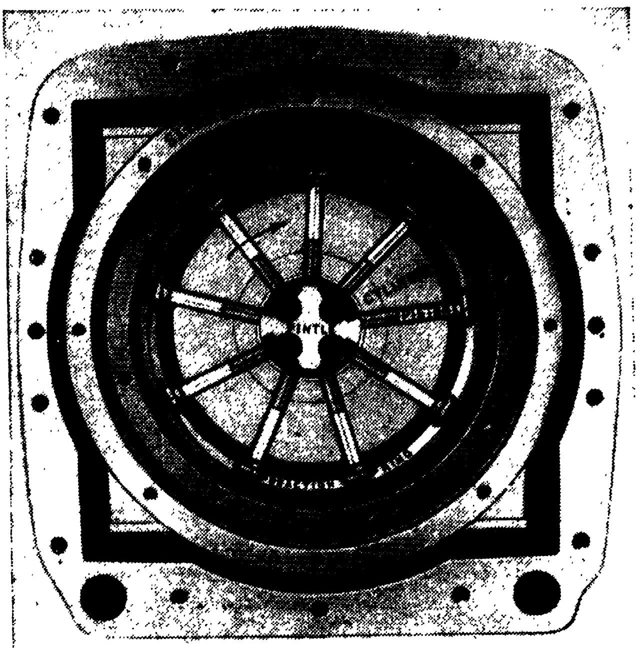


FIG. 84. Neutral position of Oilgear pump. (*The Oilgear Co., Milwaukee, Wis.*)

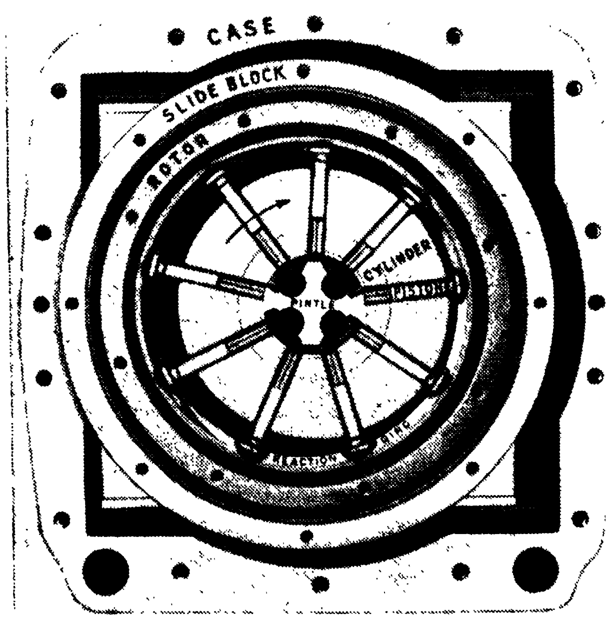


FIG. 85. Maximum-stroke position of Oilgear pump, left-hand. (*The Oilgear Co., Milwaukee, Wis.*)

reciprocating motion is so imparted to the pistons that those passing over the upper port in pintle are delivering oil to that port, while those passing over the lower port are sucking or filling up with oil. When the center lines of cylinder and rotor do not coincide, the differences between the radii from the center of cylinder to points of contact of the several piston heads with the conical reaction-ring surface in the rotor unit cause the piston heads to move faster or slower than their points of contact

with the reaction ring. This difference in speed is adjusted by slow partial rotation of each piston in its bore, in one direction during one half revolution and in the opposite direction during the other half revolution. The pistons thus rotate and reciprocate simultaneously.

As the slide-block-and-rotor unit is moved to the right of the cylinder-barrel center line by the control mechanism (see Fig. 86), reciprocating motion is so imparted to the pistons that those passing over the lower port in pintle are delivering oil to that port, while those passing over the upper port are sucking or filling up with oil. The position and movement of slide block are controlled very accurately, thus permitting

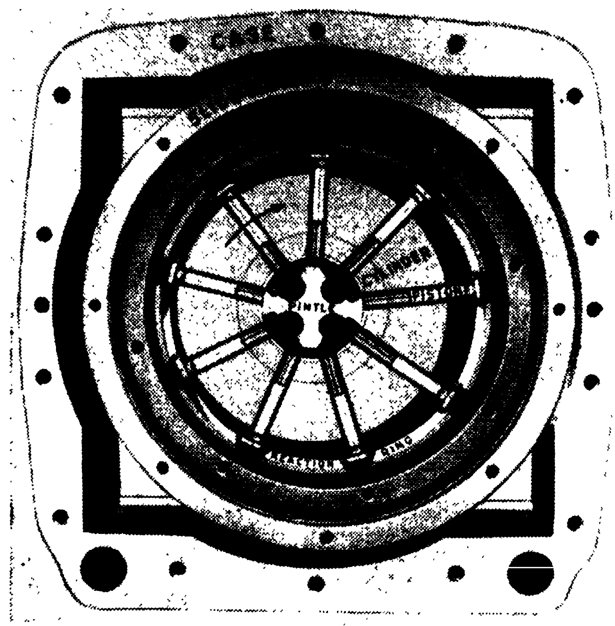


FIG. 86. Maximum-stroke position of Oilgear pump, right-hand. (*The Oilgear Co., Milwaukee, Wis.*)

the oil delivery to be varied smoothly over a stepless range in either direction from zero to maximum.

The reciprocating motion of the pistons is modified harmonic, very similar to the Hele-Shaw pump, illustrated in Fig. 74. A further modification is caused by the fact that the contact point between piston and rotor continually changes, thus changing the length of the connecting rod. The correction due to this is minute and need not be taken into consideration.

OILGEAR CONSTANT-DISPLACEMENT PUMPS. Figure 87 shows the company's constant-displacement pump. The unit operates on the same principle as the variable-displacement unit, except that the reaction-rotor bearings are mounted in the pump housing instead of in a slide block, and their center is offset a fixed amount relative to the cylinder-block center. Pumping action takes place by revolving the cylinder

barrel as in the variable-displacement unit. The pump is equipped with an external flange-type relief valve for limiting the pressure in the system. They may be furnished with or without auxiliary gear pump.

Oilgear units show excellent efficiencies over a wide range. Over-all efficiencies reach 90 per cent in their low-pressure series of pumps. Oil recommended by the manufacturer for the majority of the units when operating at temperatures above 45°F should have a viscosity of about 300 SSU at 100°F.

Three series of pumps are offered: Series 1,100 for pressures of 1,100 psi continuous to 1,350 psi peak; Series 1,700 for 1,700 continuous to 2,050 peak; and Series 2,500 for 2,500 continuous to 3,000 peak. Operating speeds are from 1,200 to 900 rpm according to size. Pumps range in capacity from 1.7 gpm for the smallest to 108 gpm for the largest unit. Both one- and two-way (reversible-discharge) variable-delivery pumps and constant-delivery pumps are available in these sizes and capacities.

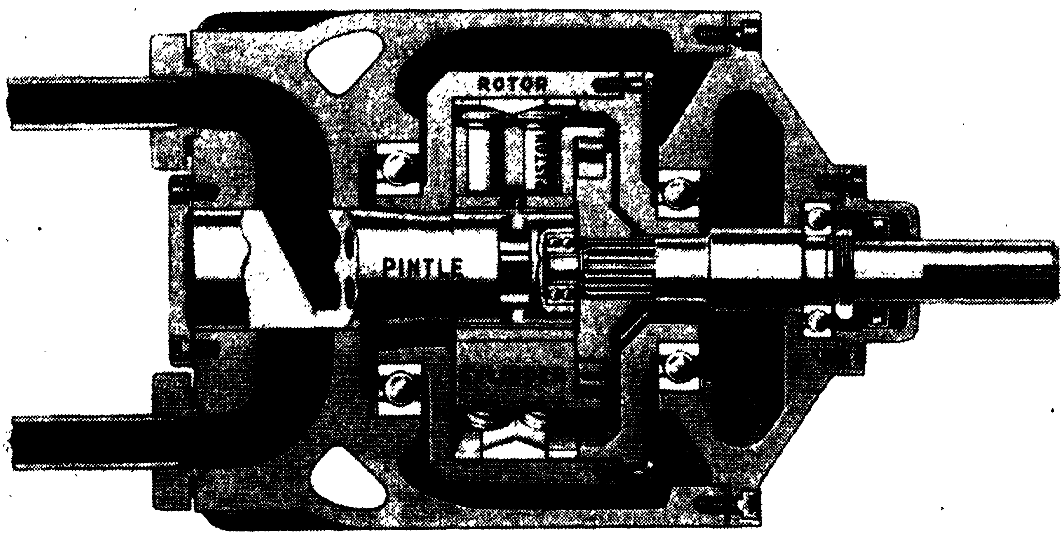


FIG. 87. Constant-delivery Oilgear pump. (*The Oilgear Co., Milwaukee, Wis.*)

The company also offers a line of one- and two-way variable- and constant-delivery duplex pumps, consisting of a radial piston pump of standard design and a large-volume low-pressure (300 psi) gear pump. These pumps are available in various sizes and pressures having normal high-pressure rating from 2 to 60 hp. The combined discharge from the gear pump and radial piston pump provides a large volume for rapid traverse at pressures up to 300 psi. When the maximum rapid-traverse pressure is reached, the gear-pump volume is automatically by-passed, and only the radial-piston-pump volume is discharged.

Both the low-pressure gear pump and the high-pressure radial-piston-type pump are compactly arranged in one case. An automatic unloading

valve for by-passing the full gear-pump volume is built into the drive-shaft housing. The combined capacity of both pumps flows in one direction and is piped into one common connection. This combined volume is usually reversed or by-passed through an external control valve. All control devices available for the radial-pump units may be supplied with this combination.

Axial Plunger Pumps

Axial plunger pumps were originally developed for use as pressure generators in connection with hydraulic transmissions and have not found the wide general-purpose applications for generating hydraulic pressure that the radial type has. As a general principle, axial plunger pumps are capable of producing large capacities at relatively low pressures and are less bulky but mechanically more complicated than the radial type. Units now offered operate with casings filled with oil, while the radial units operate with the case drained.

The outstanding representatives of this design of rotary pump are the Vickers and Waterbury units, both of which are offered as pumps and transmissions. Since hydraulic motors and transmissions will be dealt with in Chap. VIII, only pumps will be discussed in the following.

The operation and construction of this type of pump may best be explained by referring to the well-known Waterbury unit made by the Waterbury Tool Co., Waterbury, Conn. Figure 88 shows the pumping unit in sectional elevation. All working parts are enclosed within the case casting, the open or large end of which is bolted by long case bolts to the valve plate. The outer end of the case carries a bushing for the main shaft and a stuffing box to make the shaft oiltight. The case is filled with oil to lubricate the moving parts and to make up any leakage from the oil used to transmit the power. An expansion tank may be provided to supply an additional volume of oil for make-up and leakage losses and to care for differential areas in the hydraulic devices operated by the pump.

Valve Plate. The valve plate, shown in Fig. 89, has a carefully prepared surface against which the cylinder barrel rotates, and it carries two semiannular passages, which receive and discharge oil from the cylinders. As the cylinder barrel rotates, the cylinder ports pass in succession over the valve-plate ports, thus providing one port to receive oil and the other to discharge oil to the cylinder.

The valve plate is provided with two relief valves, which are usually set at 1,000 psi. These relief valves also have in them small check valves, called "replenishing valves," which will admit make-up oil from the

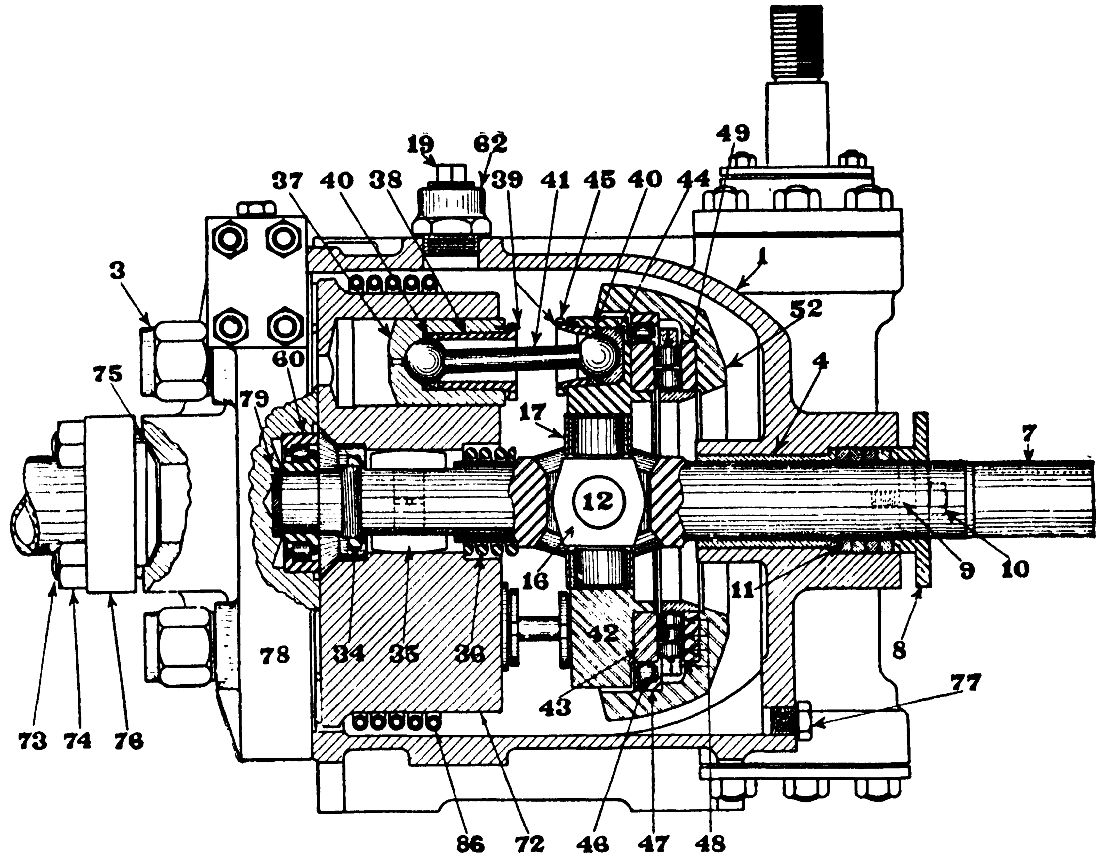


FIG. 88. The Waterbury variable-delivery pump. (*Waterbury Tool, Division of Vickers Inc., Waterbury, Conn.*)

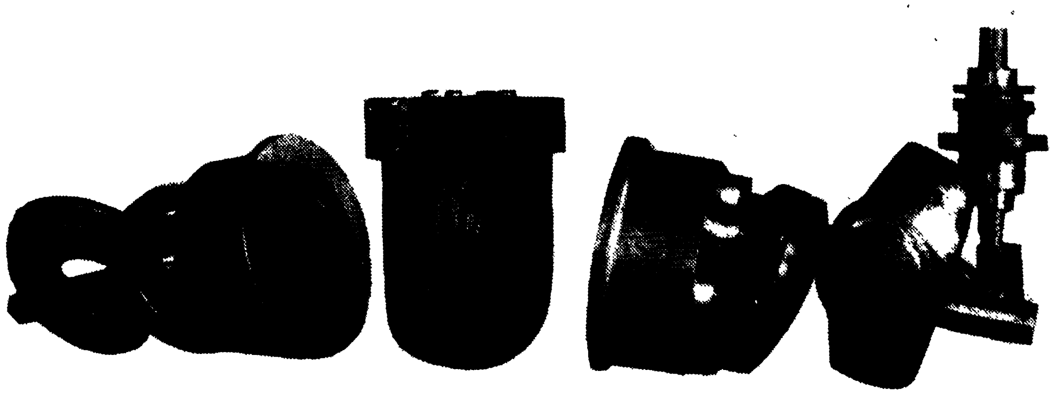


FIG. 89. Valve plate. (*Waterbury Tool, Division of Vickers Inc., Waterbury, Conn.*)

case to the low-pressure side, if needed. The valve plate also carries the outer race for the bearing supporting the main shaft.

Tilting Box. The purpose of the tilting box is to carry a thrust roller bearing against which the socket ring may rotate at the desired angle to the shaft. The tilting box is suspended and may be oscillated on two

trunnions formed on the box, which bear in bronze-bushed seats in the case. The output of the pump is governed by the angle of the tilting box. Projecting from the back of the box is a pin on which the trunnion block of the control shaft operates. The control shaft tilts the tilting box on its trunnions in either direction from the perpendicular, and this governs direction and capacity of pump output. The control consists of a threaded nut, a control shaft or screw, and suitable bearings. The

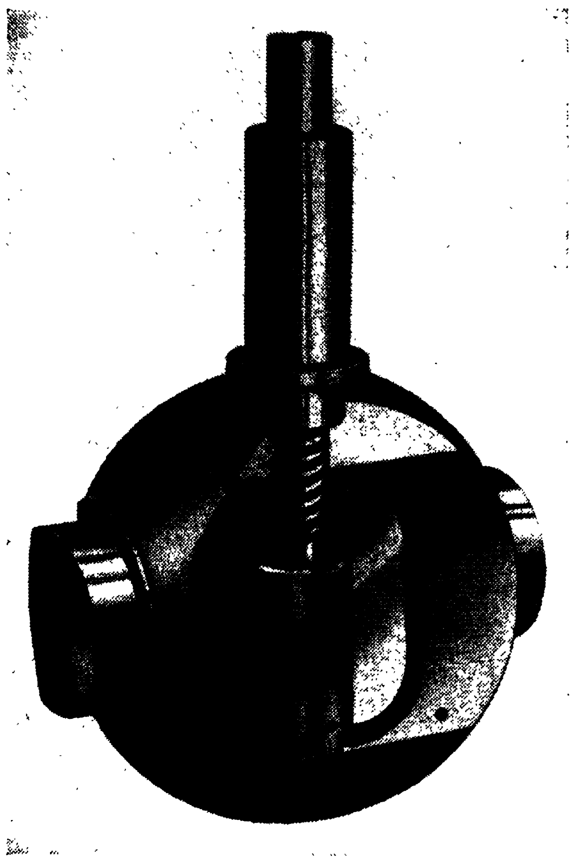


FIG. 90. Tilting box. (*Waterbury Tool, Division of Vickers Inc., Waterbury, Conn.*)

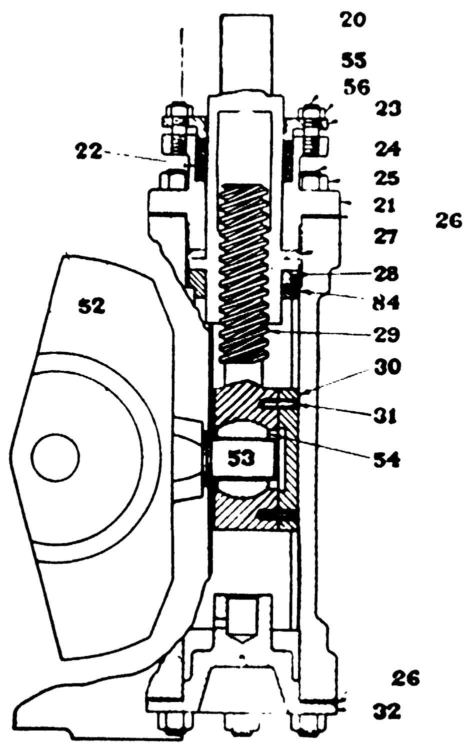


FIG. 91. Control screw. (*Waterbury Tool, Division of Vickers Inc., Waterbury, Conn.*)

threaded nut is held by its flange or collar between two thrust rings, one resting on the case and the other held by the control-nut bearing (see Fig. 91).

The threaded part of the control nut operates the screw, which is held from rotating by a spline and which carries a trunnion block in a housing on its lower end. This trunnion block can rotate in the housing and slide on the pin attached to the tilting box. As the control nut is turned, the control shaft is moved up or down, carrying with it the trunnion block, thereby changing the angle of the tilting box. Other types of control for manual or automatic actuation of the tilting box may be supplied.

Drive Shaft. The drive shaft rotates in a bushing in the case and in a roller bearing in the valve plate. Its inner end bears against the inter-shaft disk to locate it endwise and should have a reasonable play. At the socket ring, it is formed into a closed yoke to form part of the universal joint with the socket ring. The shaft carries the cylinder barrel and rotates it by two barrel keys fitting in holes in the shaft. A barrel nut is threaded on, and its only function is to prevent the barrel from slipping off when rotating parts are removed from or assembled in the case.

Cylinder Barrel. The cylinder barrel contains seven or more cylinders ground to size and is keyed to the shaft by the barrel keys. A barrel

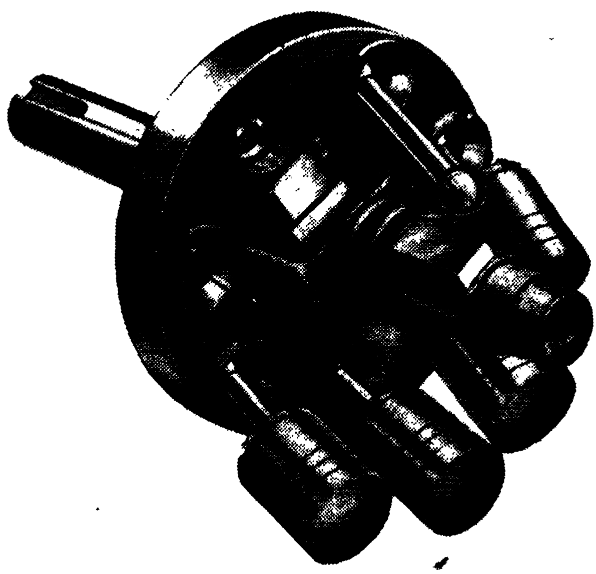


FIG. 92. Socket ring and main-shaft group. (*Waterbury Tool, Division of Vickers Inc., Waterbury, Conn.*)

spring between the back of the barrel and a collar on the shaft holds the barrel against the valve plate when not under pressure. When in operation, the cylinder barrel is held against the valve plate by the pressure on the difference in areas between the cylinder and port.

Pistons. The pistons are ground to a good working fit in the cylinder bores. Each piston is connected to the socket ring by a connecting rod having spherical ends and drilled throughout its length to admit oil from the high-pressure system to the two end bearings. The two ends of the connecting rod are held by split end-bushings and threaded cap nuts.

The socket ring is provided with seven or more bronze sockets to receive the connecting rods. On the back, the socket ring has one race of a roller bearing that transmits the thrust of the pistons. The body of the socket ring has inward projections forming pockets for the main-shaft-

trunnion bearing blocks, which are secured by taper pins. The shaft and the socket ring are connected by a universal joint consisting of a shaft-trunnioned block. This rotates through a limited angle in the bearing block in the socket ring and carries a pin that rotates in a bushing in the main shaft, making a form of Hooke's joint.

Operation. In Fig. 88, with the tilting box and socket ring set at the neutral position; that is, perpendicular to the shaft, rotation of the shaft will carry around with it the socket ring, cylinder barrel, pistons, and connecting rods, but the pistons will not reciprocate. Therefore, there is no displacement of oil. If now the control nut is turned a little to move the top of the tilting box away from the valve plate, with the shaft turning clockwise, all the pistons, as they move down on the far side of the machine, will force oil through the port in the far side of the valve plate. Likewise, all the pistons, as they move up on the near side, will slide away from the valve plate and suck oil through the port in the near side of the valve plate. The far port will thus be under pressure while the near port is in suction.

It may be readily seen that the amount of oil pumped is proportional to the tilting-box angle. Also, if the tilting box were tilted toward the valve plate, the drive shaft rotating as before, the movement of oil through the valve plate ports would be reversed.

Displacement, Power, and Efficiency. The displacement of an axial plunger pump may be computed in the same manner as that of a radial pump [see Eq. (40)]. Average displacement of all pistons is

$$Q = \frac{2eAnN}{231} \qquad \text{gpm}$$

where e = eccentricity = $R \tan \beta$

β = maximum swash-plate angle (usually 20°)

R = radius of piston circle (neglecting connecting-rod tilt)

Horsepower and efficiencies may be expressed in the same terms as shown in Eqs. (2) to (7) in this chapter. Efficiencies of this type of pump are very high. Waterbury pumps show efficiencies of 90 per cent over a range of 60 to 120 per cent of maximum load. At lower load percentages, efficiencies drop rapidly. The Vickers pump shows efficiency well over 80 per cent at as low as 20 per cent of rated load, and has volumetric efficiency of 95 per cent at 2,500 psi.

The reason for these excellent efficiencies lies in the following:

1. Minimum leakage loss due to hydraulically balanced valve plate and closely fitted pistons

2. Use of antifriction bearings and properly fitted connecting rods (no sliding friction)

Oil of about 320 SSU viscosity at 100°F is recommended by the manufacturers for this type of pumping unit. The best working temperature is about 120°F, and temperature of 155°F should not be exceeded. For continuous duty, the Waterbury unit is provided with cooling coils mounted inside the case and surrounding the cylinder rotor.

Hydraulic Working Pressure. Horizontal plunger pumps are primarily low-pressure large-displacement units. Waterbury pumps are designed for 400 psi continuous duty, with 500 psi intermittent service, and 1,000 psi starting duty.

The Waterbury units are available in the sizes shown in Table V.

TABLE V

Size no.	Speed, rpm	Working horsepower (cont.)	Capacity, cu in per min	Approx, weight, lb
2	900	8	8,000	200
5	600	12	14,000	260
10	550	24	27,000	450
20	500	40	48,000	850
50	400	80	99,000	2,000
150	250	160	170,000	4,900
172	350	300	272,100	13,300

Vickers pumps are rated at much higher working pressure, comparable with that of the radial type of pump.

Materials of Construction; Design of Pump. Horizontal plunger pumps are precision machines built from the finest materials available and with precision workmanship. Low-pressure units employ semisteel cylinder rotors and valve plates with gray-iron pistons. Connecting rods are steel forgings carburized and hardened. Bearing sockets are bronze, and bronze retainers for connecting rods are employed with steel retaining nuts. Drive shafts are steel forgings. The socket ring or swash plate is also a steel forging mounted on roller bearings for axial thrust load.

Problems encountered in the design of this type of unit may best be illustrated by proceeding with a calculation to establish the main dimensions of a unit.

Example: Let us assume that we wish to design a unit of 43-gpm capacity at a maximum pressure of 1,000 psi. Total efficiency is assumed to be 90 per cent and we have

$$HP_a = \frac{Q_0 \times 0.000583p}{e_t} = \frac{43 \times 0.000583 \times 1{,}000}{0.90} = 28 \text{ hp}$$

The geometric capacity of the pump at 900 rpm is 43/0.95 = $45\frac{1}{2}$ gpm at 95 per cent efficiency. With $Q_o = 45\frac{1}{2}$ gpm, $n = 900$ rpm, and $N = 7$, we have from Eq. (40)

$$Ae = \frac{45.5 \times 231}{2 \times 7 \times 900} = 0.84$$

To determine piston diameters and stroke, we proceed as follows: With a maximum swash-plate angle $\beta = 20°$, we have

$$e = R \tan 20° = 0.364R$$

The pistons are arranged with their centers circumferentially spaced on a circle of radius R. Assuming that 75 per cent of the circumference is occupied by cylinders, we have with seven pistons

$$d = 2R \sin \frac{0.75 \times 360}{7 \times 2} = 0.66R$$

and since $Ae = \pi d^2 e/4 = 0.84$, we have, by substitution,

$$\frac{\pi}{4} = 0.158R^3 = 0.84$$

from which follows

$$R = \sqrt[3]{6.8} = 1.895$$

and

$$d = 1\tfrac{1}{4} \qquad e = \tfrac{11}{16} \text{ in.}$$

With the displacement of the pump thus determined we proceed to compute the valve-plate slot area and the dimensions of the lap spaces. Slotted ports are provided in the bottoms of the cylinder bores to communicate with corresponding crescent-shaped slots in the valve plate. We compute the area of the ports as follows. Approximately, neglecting certain corrections that will be dealt with in Chap. VIII, the velocity of the pistons is sinusoidal, the same as shown in the case of a radial pump.

$$Q_I = A\omega e \sin \omega t$$

$$Q_{I\,\max} \text{ (for } \sin \omega t = 1) = A\omega e = \frac{Ae \times 2\pi n}{60} \qquad \text{cu in. per sec}$$

With $e = 0.6875$, $n = 900$, and $A = 1.22$, we have

$$Q_{I\,\max} = \frac{1.22 \times 0.688 \times 2\pi \times 900}{60} = 79 \text{ cu in. per sec}$$

With port slots $\frac{7}{16}$ in. wide and $1\frac{1}{4}$ in. long, the port area becomes 0.55 sq in. and the port velocity $79/0.55 \times 12 = 12$ ft per sec. This is quite high but permissible owing to the short path and to the fact that it occurs at a peak in the piston travel.

The land area on the port plate must be so dimensioned that the hydraulic pressure existing on it will approximately balance the pressure exerted against the bottom of the cylinder holes. For this, the following empirical relationships are given:

$$ANm - S(m - 1) = (d^2 - a^2)\pi \tag{41}$$

and

$$\frac{b - a}{d - c} = 0.8 \tag{42}$$

where A = area of piston
N = number of pistons
S = area of port annulus, all around
m = pressure-ratio factor
a = radius of inner edge of pressure surface
b = radius of inner edge of port annulus
c = radius of outer edge of port annulus
d = radius of outer edge of pressure surface

$m = 1.6$ to 1.8 on pumps. (On motors, $m = 1.1$ on 45° each side of center of land.) With a port-annulus width of $\frac{7}{16}$ in. and a pitch radius of $R = 1\frac{7}{8}$ in., we have

$$b = 1\tfrac{21}{32} \quad \text{and} \quad c = 2\tfrac{3}{32}$$
$$S = 13.77 - 8.62 = 5.15 \text{ sq in.}$$
$$m = 1.8$$
$$AN = 8.5$$

Therefore

$$\pi(d^2 - a^2) = 8.5 \times 1.8 - 5.15 \times 0.8 = 11.18 \text{ sq in.}$$

With $d = 2\frac{3}{8}$, we have $a = 1\frac{7}{16}$. Then

$$\frac{b - a}{d - c} = \frac{7}{9} = 0.78$$

The outside diameter of the cylinder rotor is $5\frac{1}{2}$ in. As shown in Fig. 93, the port-plate outside diameter is made to match the cylinder-rotor diameter and is subdivided into a number of segments, vented to the casing to serve as additional support for the rotor thrust. The surfaces of the segments should be slightly tapered, as in Fig. 76, to facilitate creation of a pressure-sustaining oil film.

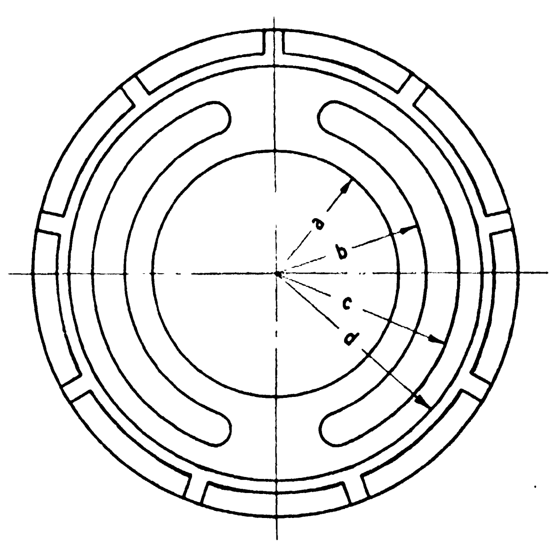


Fig. 93. Port-plate dimensions.

Suction pipe area at 5 ft per sec =

$$\frac{4.5 \times 231}{12 \times 5 \times 60} = 2.9 \text{ sq in.}$$

Two-inch pipe connections with Schedule 80 pipe will be required. Connecting-rod length L, with an L/e ratio of 4:1, equals $2\frac{3}{4}$ in. Socket diameter of connecting rods is approximately one-half piston diameter, or $\frac{5}{8}$ in.

Connecting rods are provided with axially drilled holes to permit lubrication of socket-ring bearings. These holes communicate with corresponding holes in the piston so that oil under pressure may be supplied. Bearing pressure on connecting rods is 4,000 psi. Connecting-rod diameter is approximately $\frac{5}{16}$ in. Compressive stress in connecting rods is 16,000 psi.

$$\text{Slenderness ratio} = \frac{L}{r} = \frac{2.75}{0.078} = 35$$

Critical buckling load (Rankine's formula) $p = \dfrac{S}{1 + (L^2/6{,}250r^2)} = \dfrac{100{,}000}{1 + {}^{1200}\!/\!_{6250}}$
$= 85{,}000$

$$\text{Safety factor against buckling} = \frac{85{,}000}{16{,}000} = \frac{5.3}{1}$$

Connecting rods are made from carburized and hardened alloy steel.

The socket ring is a steel forging operating on thrust and radial roller bearings. Roller bearings should be designed so that roller length does not exceed roller diameter, and arrangement and spacing of thrust rollers should be such that rollers will not "track" on race surfaces. The design of this type of bearing is a problem outside the scope of this text and need not be elaborated upon in detail. It may suffice to say that the thrust bearing should be designed for a total load of 8,500 cos β = 8,000 lb, and the radial bearing for 4,880 sin β = 1,660 lb. The socket-ring bearings are mounted in the tilting block, which is a steel casting carried on trunnions in the housing of the pump.

The drive-shaft diameter selected is 1¼ in. in bearing, 1⅛ in. at end. Torsional stress (bottom of keyway) is

$$\frac{5{,}250 \times 28 \times 12 \times 16}{\pi \times 900 \times (7/8)^3} = 15{,}000 \text{ psi}$$

The drive shaft is forked in center to receive the trunnion block of the universal joint driving the swash plate. The shaft drives the cylinder barrel by means of keys with spherical surfaces to permit a slight angular displacement of the cylinder barrel to allow for the difference in oil-film thickness between suction and discharge.

The barrel spring must have sufficient strength to permit the pistons to draw a vacuum in the cylinders without air infiltration. We calculate the spring force at a pressure of 15 psi on the net piston area of 0.67 × 7 sq in., or 70 lb. The inner end of the drive shaft is supported by a standard roller bearing of 25-mm bore and 62-mm OD.

No mention has been made in the preceding analysis of the fact that the action of the universal joint introduces a variation in the rotating speed of the swash plate, which introduces irregularities in the pump output. Ordinarily, with the pump used as pressure generator, these variations are not serious. In case of a transmission with the motor running at very slow speed, the variation becomes quite noticeable and may be compensated for by altering the circumferential or radial spacing of the cylinders. This problem will be dealt with in further detail in Chap. VIII.

The Vickers unit, described in the following, avoids this difficulty by tilting the cylinder barrel rather than the socket ring, thus maintaining uniform angular velocity of the connecting-rod sockets.

The Vickers Axial Plunger Pump. This design, originally developed by Professor Thoma in Germany, is being manufactured by Vickers Inc. in Detroit. Redesigned by Vickers engineers, the pump has been

ingeniously adapted to automotive production methods and is capable of sustaining high operating pressures at unusually high rotative speeds.

As shown in Fig. 94, the socket ring is made in one piece with the drive shaft. The drive shaft is carried in roller and ball bearings to care for both thrust and radial loads. Connecting rods are mounted in brass sockets in the socket ring and are hardened and ground steel forgings. The other end of the connecting rod is secured in the piston, which is also hardened and ground. Vickers employs bronze cylinder barrels and

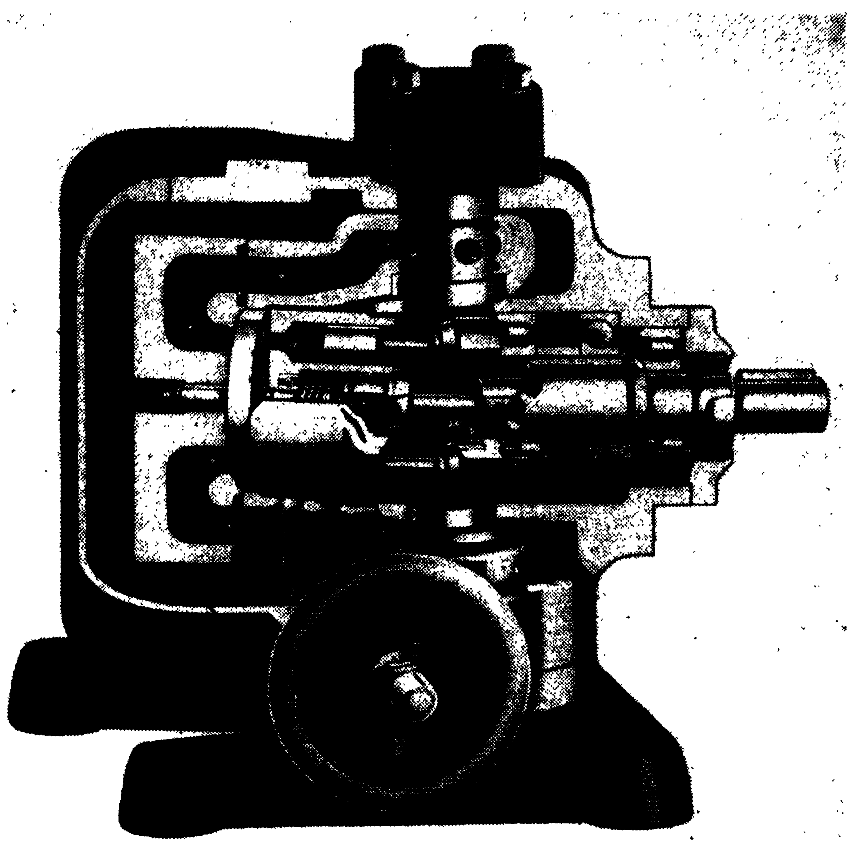


FIG. 94. Vickers axial plunger pump. (*Vickers Inc., Detroit, Mich.*)

hardened steel port plates. An ingenious arrangement permits operation at high working pressure by utilizing the maximum-sized piston rod and socket that the design can accommodate. To this end, the pistons are bored out for minimum permissible wall thickness, and instead of using retaining nuts, the pistons are merely crimped over the connecting-rod ball. Similarly, the opposite end of the rod is inserted in a brass socket, force-fitted in the socket ring, and the brass socket is crimped over. Thus the body diameter of the rod may be made as large as perhaps one-half the piston diameter, which results in manageable stresses even at pressures as high as 3,000 psi.

The barrel is driven from the drive shaft by an ingeniously designed universal joint shaft. This shaft has spherical ends with pins going through the center of the sphere, which is flattened perpendicularly to the pin axis. The pin carries two square sliding blocks, which fit into slots in a drive bushing that is inserted into the cylinder rotor and drives the rotor by means of projections engaging the slots. The cylinder barrel is carried through a ball bearing on a pin that passes through the port plate and secures the rotor to the port plate. A spring is interposed between the head of the pin and the barrel to hold the barrel to the port plate with a force sufficient to secure proper operation and maintainance of suction. Port plate is carried on a swiveling yoke, which may be

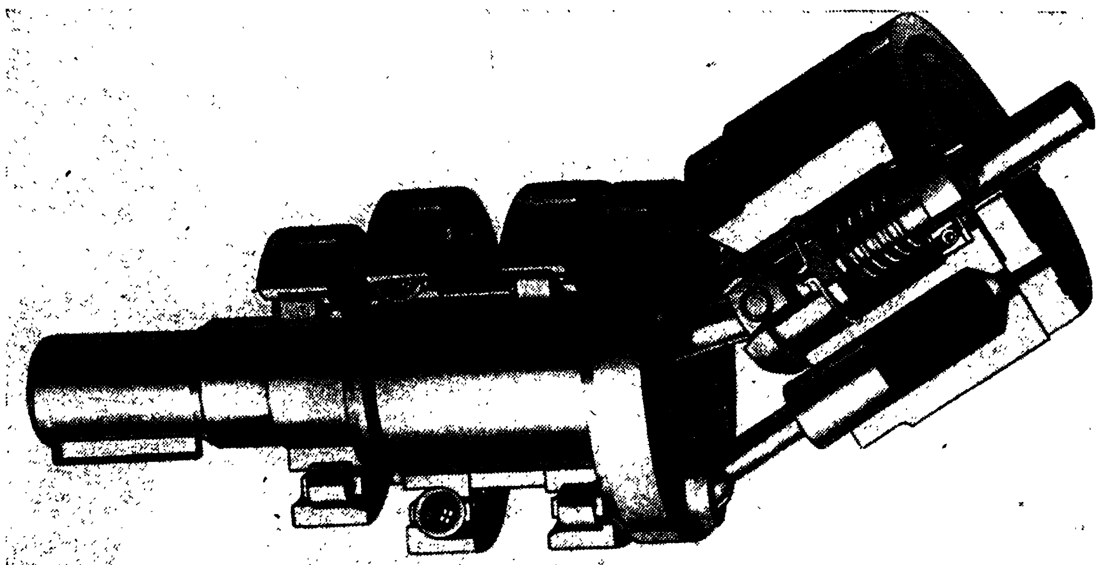


FIG. 95. Drive-shaft and cylinder-barrel arrangement. (*Vickers Inc., Detroit, Mich.*)

tilted to produce piston stroke and delivery. Passages are carried around and into the hollow trunnions that support the yoke. Pipe flanges are attached to these trunnions for suction and discharge. Figure 95 shows the arrangement of driving mechanism and cylinder barrel removed from the pump housing.

Very high rotative speeds are recommended by the manufacturer of this unit. As a result port velocities become unmanageable and supercharging must be resorted to to utilize the full rated capacity of the pumps. Vickers vane-type pumps are recommended for this purpose. The largest models are provided with power take-off for flange-mounted supercharging pumps. Approximately two-thirds of the rated delivery is obtainable with conventional suction or gravity supply.

Vickers pumps are available in capacities ranging from 5 gpm at 1,800 rpm to 340 gpm at 900 rpm. They may be equipped with any of a number of suitable controls, which will be discussed in Sec. 6 of this chap-

ter. Oil of 315 SSU viscosity at 100°F is recommended by the manufacturer, with oil temperature not to exceed 155°F.

Figure 96 shows installation dimensions of the line of pumps equipped with handwheel stroke-adjustment control.

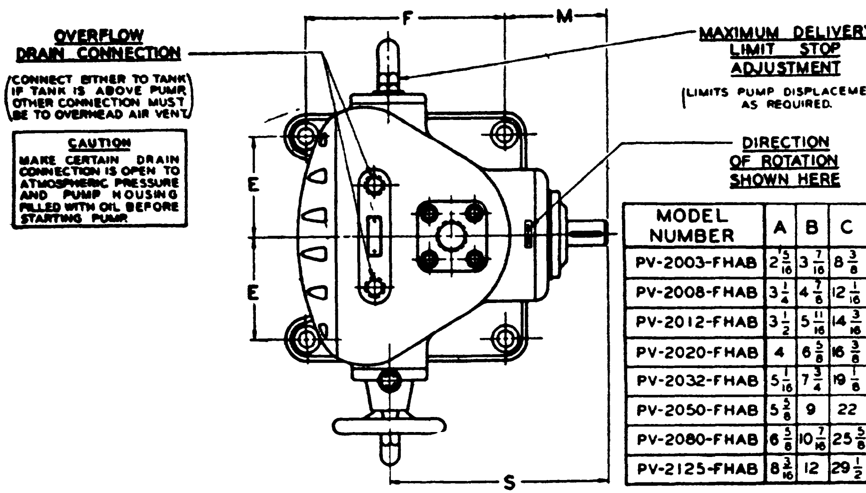
The Sundstrand PW and PWX Feed Pump. A unique axial plunger pump for machine-tool feed service has been developed by the Sundstrand Machine Tool Co., Rockford, Ill.

The unit consists of the company's Rota Roll pump, previously described, which serves as rapid-traverse and supercharging pump, together with the axial-plunger-type, variable-delivery pump, all compactly assembled in one housing. The PWX pump also contains all necessary control valves in the same housing, making a complete self-contained feed unit, requiring only a pilot valve and piping to operate the machine. The PW unit contains the pumps only, the operating and control valves being mounted separately.

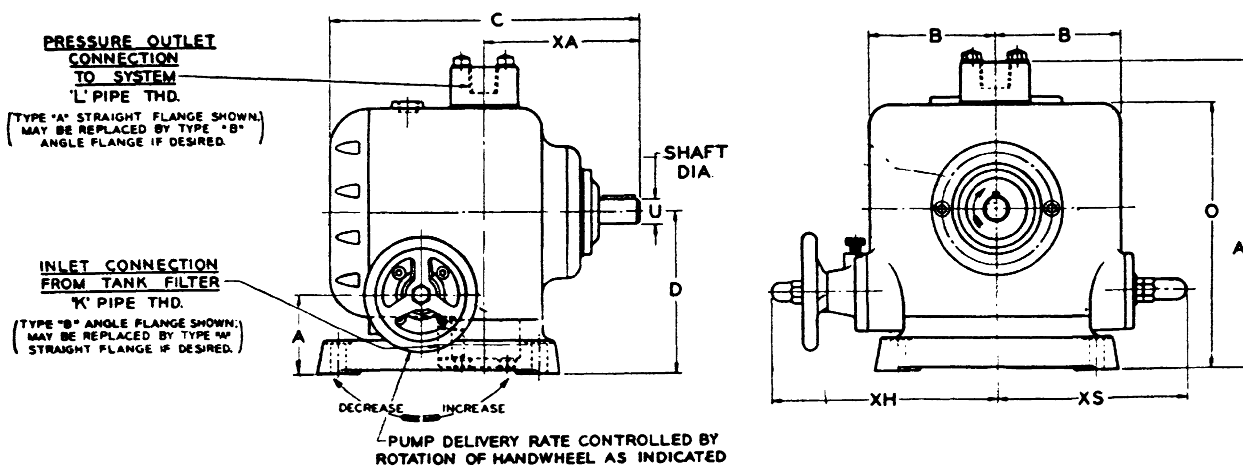
The variable-displacement or feed pump may provide one, two, or three rates of feed, all adjustable from zero to maximum. The operating principle of the pump is shown in Fig. 97. In this design, the swash-plate does not revolve, but is given a wobbling movement by the eccentric on the drive shaft. The end of the swash-plate shank carries a ball bearing with spherical outer race operating in an angular bore in the drive-shaft head. The opposite end of the swash plate is mounted on a spherical trunnion, which in turn is carried by a hydraulic piston that permits adjustment of the trunnion in axial direction. Adjustable limit stops locate the trunnion in either one of two extreme positions, resulting in two feed adjustments. It may be seen readily that the wobbling motion of the swash plate imparts reciprocating movement to the pistons, the amount of piston stroke depending upon the eccentricity of the swash-plate shank. This eccentricity in turn is altered by the axial location of the trunnion. One important characteristic of this design, which distinguishes it from all others, is that at very short strokes the clearance space between pistons and valves is at a minimum, reducing oil compression in the pump cylinders and contributing materially to the accuracy of feeding.

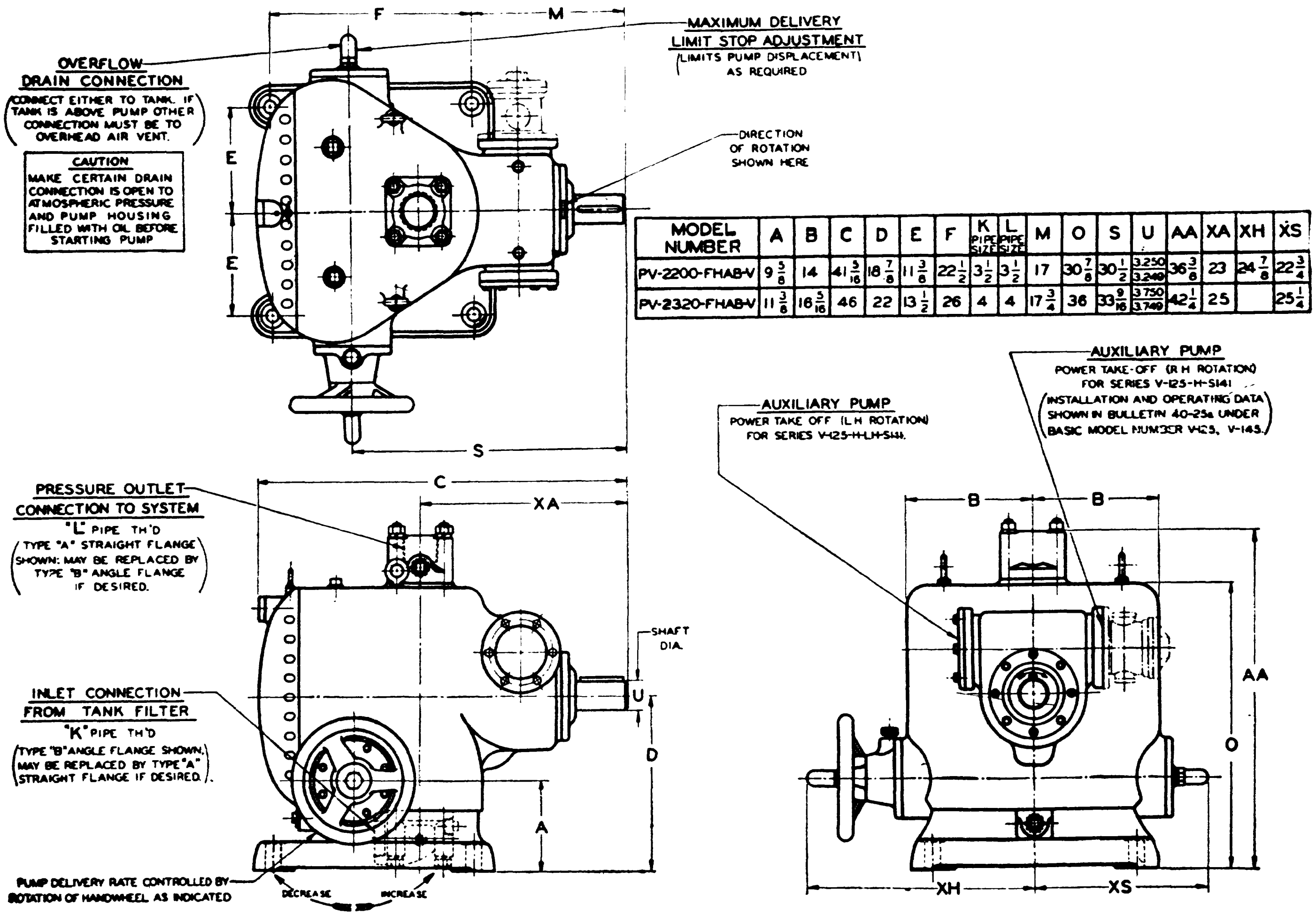
Check valves of conventional design take the place of the customarily used pintle or valve plate. The constant-displacement pump charges the piston pump, and the oil pressure keeps the pistons in contact with the swash plate. An exterior view of the unit is shown in Fig. 98.

PWX units are available in six models, with feeds in one or both directions and feed-pump capacities of 250 and 450 cu in. per min, and rapid-traverse capacity up to 4,300 cu in. per min. Operating speed for all sizes is 1,200 rpm. Pressure developed is 1,000 psi maximum for the



MODEL NUMBER	A	B	C	D	E	F	K PIPE SIZE	L PIPE SIZE	M	O	S	U	AA	XA	XH	XS
PV-2003-FHAB	$2\frac{5}{16}$	$3\frac{7}{16}$	$8\frac{3}{8}$	5	$2\frac{3}{4}$	$5\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	3	$7\frac{7}{8}$	$6\frac{5}{16}$	$\frac{.750}{.749}$	$9\frac{1}{2}$	$4\frac{3}{8}$	$7\frac{3}{8}$	$5\frac{5}{16}$
PV-2008-FHAB	$3\frac{1}{4}$	$4\frac{7}{8}$	$12\frac{1}{16}$	$6\frac{3}{8}$	4	8	1	1	$4\frac{1}{8}$	$10\frac{1}{4}$	$8\frac{3}{4}$	$\frac{.9995}{9990}$	12	$6\frac{1}{4}$	$8\frac{7}{16}$	$7\frac{1}{2}$
PV-2012-FHAB	$3\frac{1}{2}$	$5\frac{11}{16}$	$14\frac{3}{16}$	$7\frac{3}{16}$	$4\frac{1}{2}$	9	1	1	$4\frac{3}{4}$	$11\frac{3}{4}$	$10\frac{1}{8}$	$\frac{1.1245}{1.1240}$	$13\frac{5}{8}$	$7\frac{1}{8}$	$10\frac{3}{8}$	$9\frac{3}{8}$
PV-2020-FHAB	4	$6\frac{5}{8}$	$16\frac{3}{8}$	$8\frac{1}{4}$	$5\frac{1}{4}$	$10\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$5\frac{3}{8}$	$13\frac{5}{8}$	$11\frac{1}{2}$	$\frac{1.2495}{1.2490}$	$15\frac{3}{4}$	$8\frac{1}{8}$	$11\frac{1}{8}$	$10\frac{1}{8}$
PV-2032-FHAB	$5\frac{1}{16}$	$7\frac{3}{4}$	$19\frac{1}{8}$	10	$6\frac{1}{8}$	$12\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$6\frac{3}{8}$	$16\frac{1}{4}$	$13\frac{5}{8}$	$\frac{1.4995}{1.4990}$	$18\frac{11}{16}$	$9\frac{5}{8}$	$13\frac{1}{4}$	$12\frac{3}{8}$
PV-2050-FHAB	$5\frac{5}{8}$	9	22	$11\frac{3}{8}$	$7\frac{1}{8}$	$14\frac{1}{4}$	2	2	$7\frac{1}{4}$	$18\frac{11}{16}$	$15\frac{5}{8}$	$\frac{1.7495}{1.7490}$	$21\frac{13}{16}$	11	$14\frac{9}{16}$	$13\frac{11}{16}$
PV-2080-FHAB	$6\frac{5}{8}$	$10\frac{7}{16}$	$25\frac{5}{8}$	$13\frac{3}{8}$	$8\frac{5}{16}$	$16\frac{5}{8}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$8\frac{3}{8}$	22	$18\frac{1}{8}$	$\frac{1.9995}{1.9990}$	$25\frac{11}{16}$	$12\frac{3}{4}$	$16\frac{15}{16}$	$15\frac{3}{4}$
PV-2125-FHAB	$8\frac{3}{16}$	12	$29\frac{1}{2}$	$16\frac{1}{16}$	$9\frac{3}{4}$	$19\frac{1}{2}$	3	3	$9\frac{3}{8}$	26	$21\frac{1}{8}$	$\frac{2.2495}{2.2490}$	$30\frac{3}{4}$	$14\frac{1}{2}$	$18\frac{11}{16}$	$17\frac{1}{2}$





MODEL NUMBER	A	B	C	D	E	F	K PIPE SIZE	L PIPE SIZE	M	O	S	U	AA	XA	XH	XS
PV-2200-FHAB-V	$9\frac{5}{8}$	14	$41\frac{5}{16}$	$18\frac{7}{8}$	$11\frac{3}{8}$	$22\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$	17	$30\frac{7}{8}$	$30\frac{1}{2}$	$\frac{3.250}{3.249}$	$36\frac{3}{8}$	23	$24\frac{7}{8}$	$22\frac{3}{4}$
PV-2320-FHAB-V	$11\frac{3}{8}$	$16\frac{5}{16}$	46	22	$13\frac{1}{2}$	26	4	4	$17\frac{3}{4}$	36	$33\frac{9}{16}$	$\frac{3.750}{3.749}$	$42\frac{1}{4}$	25		$25\frac{1}{4}$

FIG. 96. Installation dimensions of Vickers axial plunger pumps. (*Vickers Inc., Detroit, Mich.*)

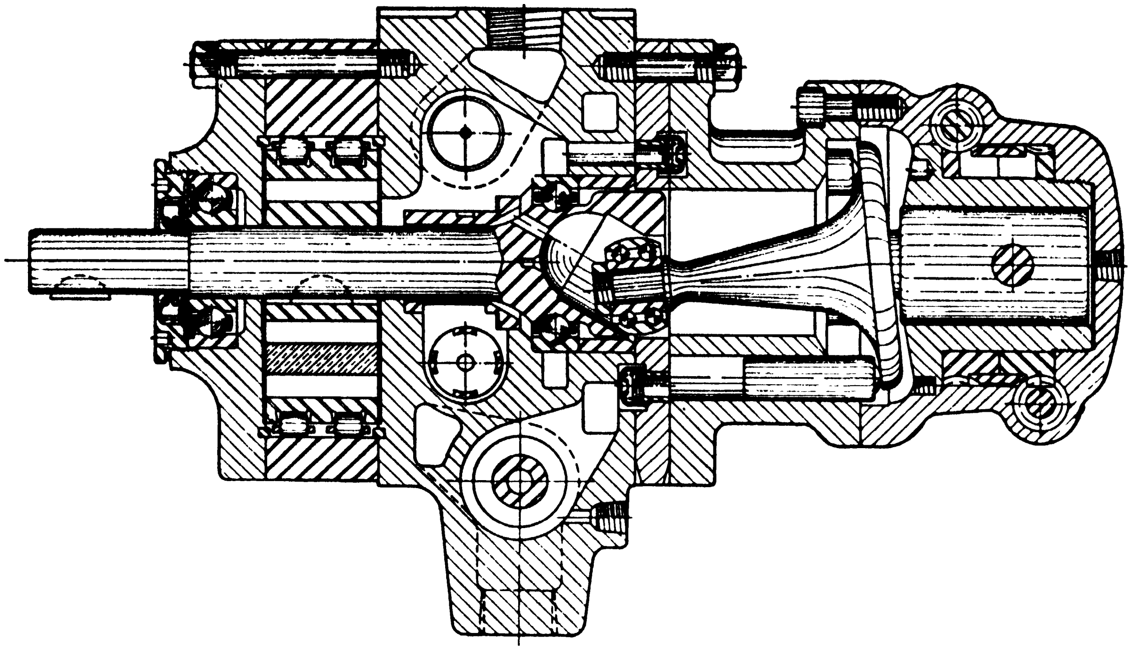


FIG. 97. Sectional view of PWX pumping unit. (*Sundstrand Machine Tool Co., Rockford, Ill.*)

FIG. 98. The PWX-series pumping unit. (*Sundstrand Machine Tool Co., Rockford, Ill.*)

piston pump and 750 psi for the rapid-traverse pump. Oil of 150 SSU viscosity at 100°F is recommended.

5. INSTALLATION AND OPERATION OF HYDRAULIC PUMPS

Operating Oil. Pumps described in the preceding sections of this chapter are precision-made pieces of equipment, constructed of the finest materials available and fitted with close clearances for efficient operation and minimum leakage. This fact should be continually kept in mind by those designing hydraulic systems and installing and using these units. Recommendations given by the manufacturers for oils to be used should be strictly adhered to. This text contains information on and requirements for oils suitable for use in these units. Only the best and highest

grades of lubricating oils should be considered for operation of these pumps.

Oil-tank Design and Capacities. Oil reservoirs may be designed to form the base for pump and motor. Either cast-iron or welded-steel construction is satisfactory. The design must provide for cleanout openings sufficiently large to permit thorough cleaning of tanks. It is essential to have all openings properly gasketed and sealed to prevent entrance of dirt and grit. The tanks must be vented to the atmosphere to permit the pump to draw oil by suction. For these vent openings, combined strainers and air vents have been designed by Vickers Inc. and others, which permit filling the oil tanks by pouring oil through a strainer, which at the same time serves as air vent. Commercial types of air filters, such as the AC or Airmaze, may also be used for this purpose. Visual indication of oil level should be provided by sight glasses or oil-level gauges. All oil lines entering the reservoir should be carried well below the oil level to prevent air infiltration. Tanks should be baffled near inlet or suction openings to prevent agitation of the oil. In some hydraulic circuits, auxiliary tanks are used that serve to prefill the hydraulic cylinder during a portion of its travel. Generally, prefill tanks operate by gravity or suction. Air vents must be large enough to permit an intake of air corresponding to the withdrawal of oil from the tanks. In some installations, this may lead to sizable units. Oil-bath-type air filters, as made by the Airmaze Corporation, Cleveland, Ohio, have been found very satisfactory for this purpose. The size of the filter connection should be about one-half that of the prefill valve.

Tank Capacities. Capacity of oil tanks used with hydraulic pumps and transmissions must be sufficient to permit operation of pumps at reasonable temperatures, prevent air infiltration, and permit escaping air to reach the surface. Tank sizes actually employed are largely empirical, depending upon the capacity of the pumps, frequency of operation, absence or presence of heat-generating devices such as chokes and metering valves, magnitude of hydraulic pressure, etc. In general, tank capacities should never be less than the capacity of the pump or pumps for 1 min. In most cases, tanks having a capacity of from twice to three times the pump capacity per minute will be found satisfactory. Installations using large operating cylinders requiring great volume changes must have tanks large enough to care for these additional requirements. Prefill tanks should have a capacity of three times the cylinder volume that they supply.

Oil Temperatures. Manufacturers' recommendations for oil temperatures are given in this text. In general, temperatures of 150°F should not be exceeded for oil hydraulic installations. Heavy pressure

systems employing radial plunger pumps are held to temperatures of 120°F and less to prevent excessive slippage at the higher temperatures. For this reason, water cooling must be resorted to in some installations. Whether water cooling should be used on any particular installation is determined by requirements, type of service, and custom. Generally, medium-pressure circuits employing rotary-type pumps at pressures of 1,000 psi and less do not require water cooling. Hydraulically operated machine tools are rarely equipped with cooling systems, although recently some heavy-duty metal-working machines have appeared employing this means of oil-temperature control. Oil hydraulic presses employing high pressures are generally equipped with water coolers. Hydraulic transmissions operating continuously under heavy load are sometimes water-cooled.

The following empirical rule may be used to determine the necessity for and the amount of cooling required. The heat that is generated in a hydraulic system may, in general, be assumed to be the equivalent of about 20 per cent of the connected horsepower. The figure may vary considerably above or below this amount, depending upon the efficiency of the pump and circuit, the existence of heat-generating devices, etc. The pumping unit, hydraulic cylinders, reservoir, and piping may serve to dissipate most or all of this amount, depending upon conditions. For medium-pressure installations with moderate service, oil tanks in accordance with the recommendations of this paragraph will generally dissipate the heat generated in the hydraulic system. A heat-transfer coefficient of about 2 Btu per °F per sq ft per hr may be used to calculate heat dissipation.

Heat dissipation by natural radiation may be assisted by installation of water cooling coils in the oil tank. This is a low-cost but not very efficient means and is economical only for small installations. Copper coils may be used for this purpose. Coils of this type will dissipate 10 to 15 Btu per hr per °F per sq ft of surface.

Heavy-duty high-pressure installations should be equipped with oil coolers and circulating pumps. Several makes of coolers are available that have given satisfactory service. Figure 99 shows the cooler made by the Harrison Radiator Co., Division of General Motors, Lockport, N.Y. The unit consists of a series of cooling plates enclosed in a cast-metal housing. The individual plate consists of an upper and lower stamping of nonferrous metal alloy, which encloses a distributor strip that breaks up the oil stream and adds to the structural strength. Several of the plates are assembled to form the cooling element or core, the plates being so spaced that the cooling water circulates freely over the exposed surfaces. The header plate is added, and the entire core assembly is

copper brazed in a deoxidized atmosphere. The assembly is mounted in a cast housing with a cast cover plate suitably gasketed to prevent water leakage. Cooling water passes through the housing. The unit will withstand reciprocating pressures up to 75 psi and constant pressure up

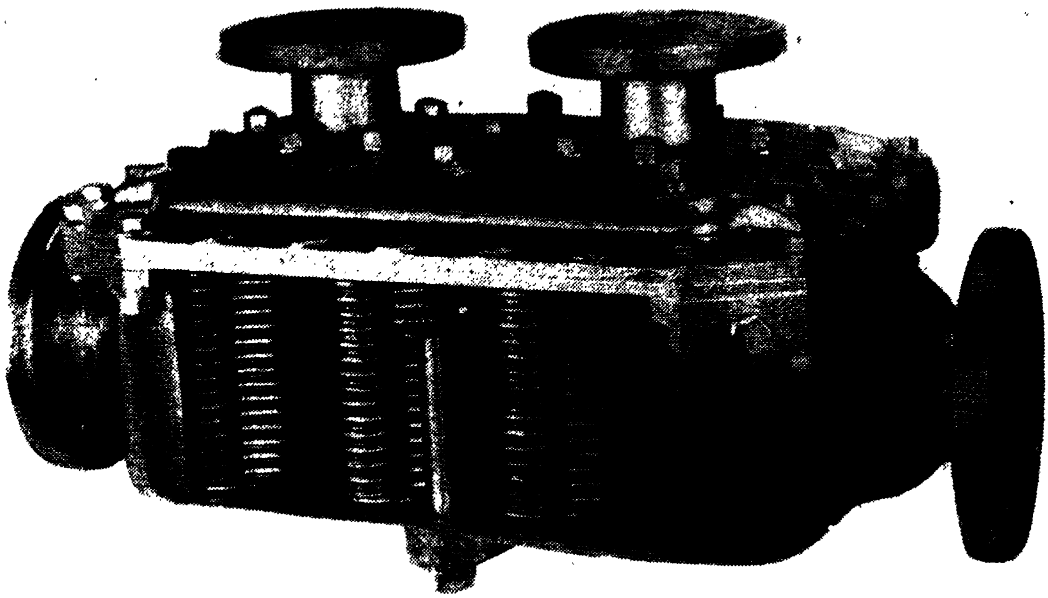


Fig. 99. Harrison plate-type oil cooler. (*Harrison Radiator Co., Lockport, N.Y.*)

to 150 psi. Harrison coolers are available in eight sizes to meet any requirement within normal operating range of hydraulic equipment.

Table VI contains recommendations for coolers and circulating-pump capacities. It must be remembered that at lower temperatures and/or

Table VI. Cooler Sizes and Circulating-pump Capacities*

Size of cooler	Oil flow, gpm	Oil pressure, psi	Oil temp. drop, °F	Capacity, Btu per min	Hp
HE 206	3½	5	10.4	128	3.0
HE 209	3½	3½	11.6	140	3.2
HE 212	3½	3½	12.4	158	3.6
HE 45	5	5	10.8	180	4.2
HE 50	10	7½	12.8	400	9.4
HE 60	15	7	16	800	18.1
HE 70	25	7½	16.8	1,400	33
HE 80	35	8	17.6	2,000	47

* Based on average conditions, and the following data:
Oil viscosity, 350 to 500 SSU at 100°F
Oil temperature entering cooler, 130°F
Cooling-water-to-oil flow ratio, 2:1
Cooling-water temperature, 75°F.

with more viscous oils much higher pressures are developed in these coolers. Circulating pumps, if separately driven, should not be undermotored for this reason, and care must be used not to exceed the pressure rating of the coolers, especially in reciprocating service.

Most radial- and axial-plunger-type pumps are equipped with auxiliary pumps that may be used to circulate oil through the cooler. Tapped holes are provided for this purpose on the Oilgear unit to take off oil flow for coolers and pipe it back into the pump housing. Similarly, gear pilot pumps supplied with the Hydro-Power units may be utilized for this purpose. If we assume that the horsepower removed by cooling constitutes one-fifth of the total connected horsepower, and if we figure roughly 1 gal high-pressure output per connected horsepower, then the circulating-pump capacity should be an average of 20 per cent of the main-pump capacity. This will result in sufficient capacity for efficient operation of auxiliary hydraulic controls, servo motors, etc., as will be explained later.

Another very satisfactory cooler is the Ross cooler, made by the Ross Heater and Mfg. Co., Buffalo, N.Y. This unit is of the conventional multiple-tube type, with tubes mounted in a housing between headers. In the Ross cooler the water passes through the tubes and the oil around them. Units are available of almost any capacity to meet any required condition. Circulating oil capacities correspond to those worked out for the Harrison coolers.

Oil cooling may be made automatic by the use of any of the number of thermostatically controlled valves on the market. The author prefers for this purpose electrically operated valves controlled by thermostatically actuated switches.

Cleanliness. It will hardly be necessary to convince the reader who has attentively followed the descriptions and instructions of this text that utmost cleanliness and care in excluding dirt, grit, and foreign matter is one of the essentials for the construction and operation of hydraulic units. To this end, the greatest care must be used by the builder and user of hydraulic units to exclude these enemies of a hydraulic system. All castings should be pickled before machining to remove core sand and carefully washed, and burrs, chips, and shavings must be removed after machining. Oil-resisting paint should be applied to all inside surfaces that are not finished. Heat-treated parts should be sandblasted to remove scale. All machined parts should be degreased and treated with dry film before being placed in stock. Assembly should proceed with utmost cleanliness, and good housekeeping should prevail in the assembly shop.

Similar precautions must be used in testing the units. Oil supply

used in testing should be periodically filtered and renewed occasionally. Testing of the units may be preceded with a reasonable run-in period, gradually increasing pressure and/or speed until the rating is reached. Temperature and power consumption should be checked and efficiency of the unit determined.

Before shipment all openings should be hermetically sealed. Plastic pipe plugs are now available for this purpose that make a tight seal excluding dirt and water and will not be mistaken for parts of the unit.

An equal responsibility for maintaining care and cleanliness rests upon the users. When installing the units, care must be taken that the reservoirs, tanks, and parts of the system are absolutely clean. Pipes should be pickled and examined for scale and residue. When repairs are necessary on a hydraulic system, the workmen should be cautioned not to track dirt and shavings off the floor into oil tanks or leave openings uncovered in dirty and dusty atmosphere.

Some hydraulic systems are equipped with pressure oil filters operated by the circulating pumps. These filters have merit where the amount of oil in use is small, as in aircraft hydraulic systems and small machine tools. In large hydraulic systems containing several hundred gallons of oil, the author has found them of little value. The author does not recommend suction-type oil filters. A good pressure filter for hydraulic work is made by the Cuno Engineering Company of Meriden, Conn.

Loads and Speeds. It should be almost unnecessary to point out that the manufacturer's recommendations for operating speeds and pressures should be rigorously observed. The temptation seems to be great to get just the extra few hundred pounds or rpm, to use the next smaller unit, to lower the cost, etc. Such practices will reap a bitter reward in premature wear, maintainance trouble, and loss of customer good will. Most standard units now on the market will give excellent service when operated within their rating and contain reasonable margins for operational overloads, but the designer should never attempt to borrow these margins on his original specifications.

Great care should be taken to have suction lines of proper size and without restrictions. Manufacturers' recommendations should be followed on the size of these lines. In general, suction lifts over 3 ft are not recommended.

The preceding recommendations apply to all makes of pumps. For the operation of variable- and reversible-delivery plunger pumps the following is of importance. Variable-delivery pumps may be used as one-way pumps with operating valves to reverse the flow to the operating cylinder, or as two-way pumps where the reversal of flow is accomplished by reversing the pump. With the former hookup, the flow through the

pump is always in one direction. In this case, the pumps may be mounted on reservoirs and supplied with oil by suction. The discharge from the operating valve should be returned to the tank at a point relatively remote from the location of the suction pipe or separated from it by a baffle. In radial pumps, which operate with the case drained, leakage or slippage drains from the bottom of the casing back into the oil tank. For this purpose a tapped hole is provided in the bottom of the pump casing. A pipe nipple may be screwed into this hole and terminates above the oil level. If this nipple passes through a clearance hole in the tank cover, it should be sealed to prevent entrance of dirt. Horizontal plunger pumps operate with the case filled and are provided with overflow vents. Pumps may also be supplied with suction oil from overhead tanks. In this case, radial pumps are provided with a slippage or leakage tank from which the slippage is removed by a scavenging pump and returned to the overhead tank. Horizontal plunger pumps do not require this provision, but are vented to the overhead tank. One-way pumps may be connected in a closed circuit by connecting the return line from the operating valve to the suction connection of the pump and supercharging the entire system by means of an auxiliary rotary pump. Differential oil between both sides of the actuated cylinder or hydraulic motor is made up by the supercharging pump, or permitted to escape through the auxiliary-pump relief valve. Generally a check or foot valve in combination with auxiliary suction pipe is installed to ensure an oil supply to the plunger pump independent of supercharging. Supercharging pressures range from 35 to 65 psi.

In two-way, or reversing, pump systems we again find atmospheric and supercharged systems. One of the characteristics of two-way pump operation is that the oil is continually shuttled back and forth between both sides of the system, which makes it difficult to prevent air accumulation without keeping a positive head of oil on the entire system. On nonsupercharged systems, therefore, pump and operated cylinder or motor are often mounted below the oil tank to provide a measure of force feeding of the oil. In that case, leakage drain from radial pumps is returned to the overhead tank by an auxiliary scavenging pump.

In the application of supercharged systems, it should be remembered that the supercharging pump must make up the volume due to the difference in area between both sides of the hydraulic cylinder operated by the pump. Supercharging pumps built into plunger-pump units or driven by power take-offs may not have sufficient capacity to supply this volume in case of extremely large differentials, such as are often encountered in hydraulic-press work. In such cases, recourse must be taken to auxiliary atmospheric suction or extra-large supercharging pumps. Great care is

indicated, when employing atmospheric suction, to prevent air accumulation at high points in the system. Hydraulic cylinders with overhead prefill tanks and prefill valves mounted at the highest point of the cylinder are self-purging.

To summarize, the designer should remember that the four enemies of an oil hydraulic system are

1. Dirt
2. Heat
3. Water
4. Air

Determination of Motor Size. The actual horsepower required to operate a pump at a given pressure may be computed with the aid of the formulas given in preceding sections or taken from information or tables supplied by the manufacturer. Care should be taken to include power consumption of auxiliary gear pumps, power-take-off devices, etc. With the power for the peak loads determined, the size of motor may be determined by computation of the rms horsepower.

$$\text{Rms horsepower} = \sqrt{\frac{P_1}{100} H_1^2 + \frac{P_2}{100} H_2^2 + \cdots + \frac{P_n}{100} H_n^2} \qquad (43)$$

where $P_1, P_2, \ldots, P_n$ is the percentage of time during which the horsepowers $H_1, H_2, \ldots, H_n$ are applied. Of course,

$$P_1 + P_2 + \cdots + P_n = 100.$$

Rating of motor selected should not be less than the computed rms horsepower. The motor thus determined must be checked for pull-out torque. As a safe rule, the maximum torque required to operate the pump at the heaviest loading should not exceed the normal rated running torque of the motor by more than 100 per cent. The pull-out torques of most commercially available electric motors are safely above this figure. Caution is indicated where special motors are used, and motor manufacturers should be consulted.

In general, belt or chain drives are not recommended for hydraulic pumps, as no provisions are made for the bending loads imposed upon the shafts. Some manufacturers permit operation of their pumps at reduced pressures under these conditions. In all other cases, outboard bearings must be resorted to, with pulleys mounted between two bearings and pulley shaft coupled to pump shaft by flexible coupling. Connection between motor and pump shaft is made by flexible coupling. Care should be taken to align pumps and motors properly to prevent vibration and stresses.

Almost all hydraulic pumps accompany their work with a certain amount of noise. This is less pronounced in small low-pressure units but may become quite obnoxious in large high-pressure pumps. Vane pumps are the quietest running of the rotary type. Noise in rotary plunger types may run from a slight whine to a pronounced humming or buzzing sound typical for this type of unit. Provision of adequate ample suction and freedom from entrained air will assist in reducing this noise. Restriction in suction lines will produce loud rattling and hammering noises. The author has found that abatement of the noise will be facilitated by mounting units on subplates of heavy and substantial section, which in turn are carried on rubber mountings such as those made by U.S. Rubber Co. or Firestone. Natural frequencies of most rubber mountings in their recommended deflection range are sufficiently low to assure almost complete insulation from the rest of the machine, leaving only air- and pipe-borne noises.

6. VARIABLE- AND REVERSIBLE-DELIVERY-PUMP CONTROLS

As previously shown, all variable-delivery pumps have sliding or oscillating members to adjust the eccentricity of the pumping action and thus vary the delivery. In order to produce adjustment of these members, a number of controls has been developed by the builders of these pumps, which may be applied interchangeably to their pumps to produce shifting and adjustment movement, maintain given pressure and stroke settings, etc. In the following, the basic controls available will be described in the different forms in which they are embodied by the respective manufacturers, and design information on some of the most important will be given.

Stem Control. This control merely consists of a sliding stem or pin, attached to the pump-shifting means, to which an external mechanism may be attached to vary the volume or reverse the discharge. This type of control is available for Vickers pumps and may also be supplied, in modified form, with the Hele-Shaw pump, as shown in Fig. 77. In this illustration, two stems are shown threaded in the pump shift rings, having a yoke or crossbar attached to them. By connecting suitable actuating mechanism to this crossbar, the position of the pump shift ring may be adjusted. The force required to move the pump-stroke shift ring or maintain it in its set position is, of course, of interest to the designer of controls. In a radial pump having harmonic motion, the horizontal force opposing or assisting stroke movement at any angle ωt will be

$$F = pA \sum_{i=1}^{i=N} \cos\left(\omega t + \frac{2\pi i}{N}\right) \tag{44}$$

where A = area of one plunger
p = working pressure
N = number of pistons

If the summation is carried out for angles ωt of 0 to 180°, it will be found that the force fluctuates between $+0.5pA$ and $-0.5pA$. The total of all fluctuations cancels out over the arc of 180°. For purposes of control design, a force $F = 0.5pA$ to $0.75pA$ should be used to deal with the peak variations. Mechanical or hydraulic actuating devices should be designed to furnish this force.

Handwheel Control. A stem extension from the stroke-control crosshead is threaded to receive a nut onto which is keyed a handwheel, by means of which the location of the stroke crosshead may be adjusted. Figure 83 illustrates the handwheel control of the Oilgear pump. The springs shown on the end opposite the handwheel serve to eliminate backlash and permit accurate stroke adjustment. Similar controls are supplied by the manufacturers of other pumps. Ordinarily, these controls serve as stroke adjustments for one-way delivery, but modifications may be supplied for reversible delivery. The control locks the pump on a fixed predetermined delivery, and relief valves are necessary to protect the pump against overload.

Remote Adjustment of Stroke by Electric Motor. Instead of the handwheel used on the control screws, a gear reduction motor may be attached to the pump to operate the adjusting nut. Horsepower of the motor depends on torque required, which may be computed from the force required to shift the pump, developed under Stem Control above, and the speed desired for the adjustment. Care must be taken to avoid sudden stalling of the adjusting motor at the limits of the adjustment. This is done by suitable limit switches. Adjustable limit switches may be provided to select automatically any preset displacement. Protection of the pump by relief valves is necessary.

Stroke Adjustment and Reversal by Hydraulic Cylinders. Several modifications of this type of control are possible. One-way delivery with two selected outputs may be produced by small hydraulic cylinders flanged to the side of the pump casing together with adjustable stops to limit pump stroke to the desired value. Oil pressure from an auxiliary source, such as a built-in gear pump, is directed to either one of the two cylinders to produce either one of the two preselected deliveries. Stops may have setscrews, knurled-wheel, or handwheel adjustments. This type of control is particularly adapted for reciprocating machine tools to produce rapid traverse and feed. The control may be arranged for one-way action with two selected speeds or two-way action with one selected speed in each direction.

All these combinations of pump settings are accomplished by hydraulically actuated pistons in combinations with adjustable stops and pilot valves to direct the application of pilot pressure to the selected cylinders. Figure 100 shows the typical arrangement of a two-speed control as applied to a one-way pump. Cylinder bores are provided in both ends of the housing in which the actuating pistons are mounted. Pressure may be supplied to these pistons through pipe connections as shown.

The screw adjustments made with the external threaded knobs limit the stroke of the pump in the following manner: Assuming that discharge of the pump is produced by shifting the crosshead to the left in the plane of the picture, then the left-hand screw adjustment determines the maximum-stroke setting, which is produced by pilot pressure forcing the

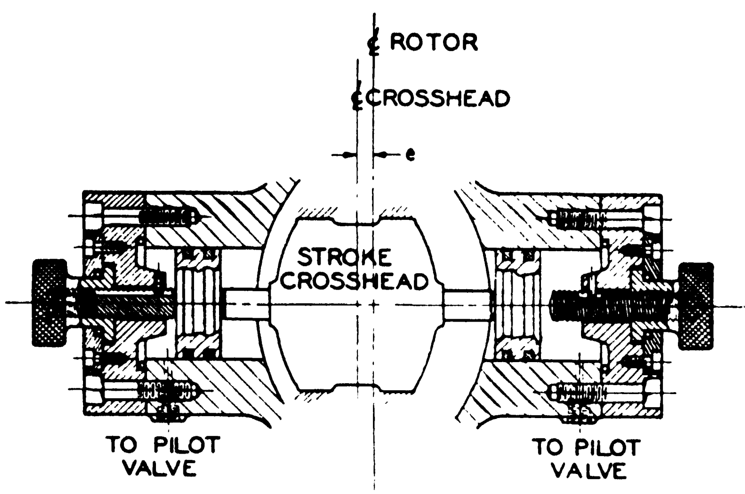


Fig. 100. Hydraulic stroke-adjustment control.

right-hand piston toward the left and against the stop. The maximum-stroke setting may be varied by changing the adjustment of the left-hand screw.

To obtain the smaller one of the two deliveries, pilot pressure is supplied to the left-hand piston, forcing the pump crosshead toward the right against the right-hand adjustable stop. This results in the minimum delivery. For one-way pump operation, the right-hand stop is so arranged that the pump cannot be shifted past the neutral position, and the result is two remote preselected volumes being available for one-way flow. If the right-hand stop is so arranged that the pump may be shifted past neutral to the right side of its delivery position, delivery in the opposite direction will take place, and we have a two-way pump with a preset delivery in each direction.

Figure 101 shows the arrangement with three positions for a two-way pump having a neutral and two adjustable end positions. By applying pressure to both sides of the controls, the pump crosshead will be forced to

neutral. Exhausting pressure at either end causes corresponding movement of the crosshead and resulting delivery. Auxiliary pistons may be provided to select additional positions for coarse or fine feeds or rapid traverse. Design and description of suitable control valves will be covered in Chap. X.

Area of control pistons must be made large enough to ensure shifting of the crosshead against the resistance of the hydraulic pressure in the pump, as covered under Stem Control. Control piston area should be equal to $0.5Ap/p_a$ as a minimum, where A is the area of one pump piston, p is the maximum pump pressure, and p_a is the pressure available in the auxiliary or pilot system. Pilot pressures customarily used vary from 150 to 500 psi.

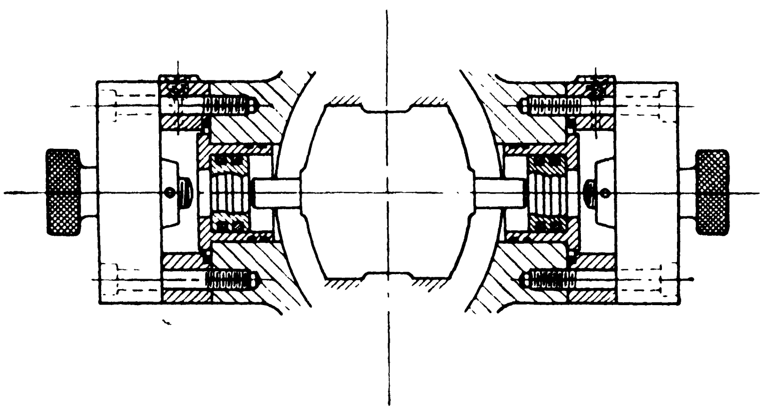


FIG. 101. Hydraulic stroke-adjustment controls with neutral position.

Oilgear units may be supplied with modifications of this control, consisting of complete machine-tool cycling devices with five positions, as follows:

Rapid traverse forward
Coarse feed forward
Fine feed forward
Neutral
Rapid traverse reverse

Selection of these five positions is made by means of a pilot valve mounted in the control unit. The pilot valve is actuated by cams or trip dogs from the carriage of the machine tool. Coarse and fine reverse feeds can also be obtained with standard units. Another Oilgear control provides selection of three positions for a reversing pump, consisting of forward, neutral, and reverse positions. All three are controlled by an electric-solenoid-actuated pilot valve. Energizing one solenoid produces forward delivery; energizing the other produces reverse. Deenergizing both solenoids causes pump to assume a neutral or no-delivery position.

Pressure-compensating Control. One of the most popular of the many control devices available for variable-delivery pumps is the pressure-compensating or -maintaining control. The basic purpose of this control is the application of a sustaining pressure by the pump at a minimum consumption of power. The application of this principle constitutes one of the most important advantages of the variable-delivery pump, which permits maintaining a preset pressure in a press or other hydraulic machine with only enough delivery of pressing fluid to compensate for leakage losses in pump and system and following up a possible yielding in the work being processed in the machine.

The operating principle of this control is basically the same for the different makes offered. It comprises a source of fixed pressure, such as a spring or hydraulic cylinder, which forces the pump crosshead on delivery stroke, opposed by a hydraulic piston that receives a supply of pressure from the discharge connection of the pump and counteracts this tendency. Discharge pressure building up in pump and system is transmitted to this piston, gradually overcoming the resistance of the fixed-pressure means and forcing the pump toward neutral or no-delivery position. The two tendencies will balance at a point where just enough delivery is produced by the pump to maintain the system pressure against the fixed resistance.

The difference in design between the different makes of control consists largely in the means used to produce and vary the fixed resistance for adjustable pressures. This may be done by springs with adjustable tension, by springs with fixed tension opposed by hydraulic cylinder with adjustable pressure, and by hydraulic cylinders.

Characteristics and operation of this type of control may best be illustrated by a sample calculation.

Example: We assume that we wish to design a pressure-compensating control for the 20-gpm radial pump designed in Sec. 4 of this chapter. The area A of one piston was 1.1 sq in. Pressure = 2,500 psi. Total control force $= F = 0.75 \times 1.1 \times 2{,}500 = 2{,}050$ lb.

$$\text{Area of control piston} = {}^{2050}\!/_{2500} = 0.82 \text{ sq in.}$$

or approximately 1-in. diameter. Spring-pressure maximum with 1-in. plunger will be

$$0.785 \times 2{,}500 = 1{,}960 \text{ lb}$$

For further computation of spring characteristics a decision must be made to what pressure the full output of the pump is to be carried until a reduction in delivery should take place. This depends on a number of considerations. Figure 102 shows the pressure-volume relationship of a spring-plunger compensating control. The length of the horizontal line at the 20-gpm or maximum-capacity point indicates the pressure up to which full discharge may be maintained. At the peak pressure the delivery is zero. The diagram contains two extremes that are impractical or impossible. One is a zero

cutoff; that is, the pump would begin reducing stroke as soon as pressure started to rise. This condition cannot be realized in practice, as no initial spring pressure would be available to put and maintain the pump on stroke. The other is maximum-pressure cutoff, which means that the pump would maintain full discharge up to the pressure limit and then cut off sharply to zero delivery. This condition is impossible of achievement with a spring-loaded control of this design, as it would require an infinitely long spring. It may be accomplished with some other types of controls, as will be described later. The general run of applications lies in the middle between these extremes, with full discharge maintained at from 40 to 80 per cent of maximum pressure. By referring to Fig. 102, it may be seen also that the cutoff pressure affects the minimum holding pressure, which cannot be less than the difference between cutoff and zero-delivery pressure. High cutoff pressures, therefore, permit greater choice in holding pressures and enable the pump to deliver full volume at a higher pressure, while lower cutoff pressures permit use of smaller motors. Motor size for pumps equipped with this control are figured with the cutoff pressure as maximum pressure

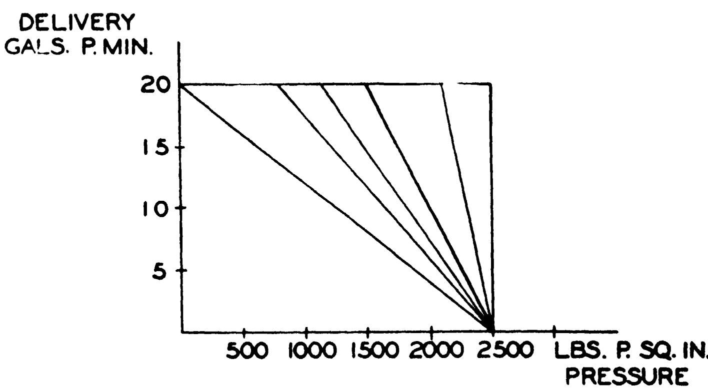


FIG. 102. Diagram of spring-plunger compensating control.

at full delivery, which may result in a material saving in motoring these pumps and in power consumption.

Example: With the above explanation, let us assume that we wish to maintain full discharge up to 1,750 psi, or 70 per cent of maximum pressure. Then, while the pump moves from full stroke to zero delivery, or $\frac{3}{8}$ in., the spring force increases from 1,750 to 2,500 psi, or $750 \times 0.785 = 588$ lb. The spring gradient, therefore, is 588/0.375 or 1,570 lb per in. The spring specification, therefore, reads 1,960 lb maximum pressure with a gradient of 1,570 lb per in.

A spring with 4-in. mean coil diameter and $\frac{5}{8}$-in. wire diameter is selected, having a stress (uncorrected) of 80,000 psi at the maximum load of 1,960 lb. The spring index is $4/0.625 = 6\frac{1}{2}$, and the Wahl correction factor is 1.22, resulting in a corrected stress of 97,500 psi.

Alloy steel SAE 6150 should be used for this spring. As a general consideration, alloy steel is recommended for control springs that carry heavy loads in order to stay within reasonable space limits.

For a gradient of 1,570 lb per in., about $2\frac{1}{4}$ acting coils are required, and we should have a total of $4\frac{1}{4}$ coils. The total compression to obtain 1,960 lb should be $\frac{1960}{1570} = 1\frac{1}{4}$ in. With an allowance of about 0.6 in. for coil clearance, we then have a free length of $4\frac{1}{2}$ in.

The general arrangement of a control designed with these specifications is shown in Fig. 103. The purpose of the choke check arrangement on the hydraulic cylinder

is to permit free access of hydraulic pressure to the control cylinder through the ball check valve to actuate the control rapidly upon the building up of pressure. In case of sudden release of pressure, the choke valve prevents slamming of the control to full stroke. For this size of control, ¼-in. pilot line is recommended.

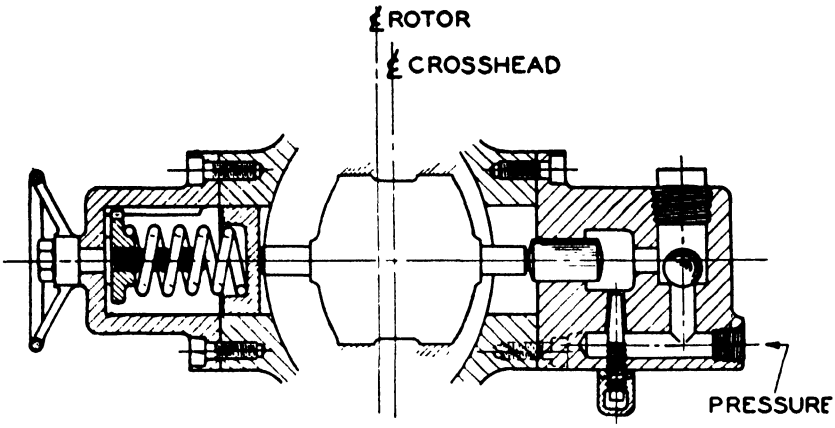


FIG. 103. Spring-plunger compensating control.

A modification of the spring-opposed plunger control is made with a fixed spring tension and adjustable hydraulic pressure opposing it. This type of control permits practically maximum-pressure cutoff, maintaining full pump delivery up to the maximum set pressure. Figure 104 shows the principle of this control. Assuming that the spring in our example above is set at a fixed pressure of 1,960 lb, we may oppose it with a piston of sufficient area to produce the minimum desired holding pressure. At 500 psi, for instance, we will require 3.9 sq in. to do this. The adjustable relief valve will then be set for any desired pressure above this minimum and will open at the set pressure to admit pressure to the control piston to force the pump to zero stroke. The pump will automatically maintain the set pressure. If there is a tendency for this pressure to drop, the relief valve will close and bleed off oil from the control cylinder to make up sufficient pump stroke to restore the equilibrium.

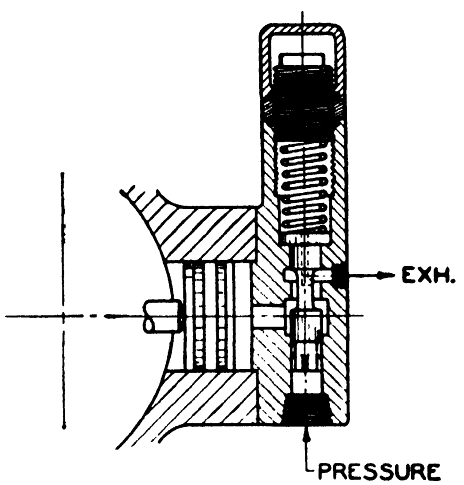


FIG. 104. Compensating control with hydraulic-pressure adjustment.

This type of control is made by the Oilgear Company. Figure 105 shows the Oilgear pump with compensating control as applied to a pressure-storage vessel. This constitutes an ideal application of the Oilgear type of compensating control, as it permits charging the pressure vessel at substantially full pump delivery just slightly below the shutoff

pressure and automatically unloading pump and motor for minimum power consumption when the maximum pressure is reached and the pressure vessel is fully charged.

In another form of control, the hydraulic piston is held between two pressures; one end opposing the spring is directly subjected to the line pressure, which bleeds through a small restriction into the opposite end, which is equipped with a relief valve of conventional design. This relief valve establishes the setting of the control. If the pressure in the line

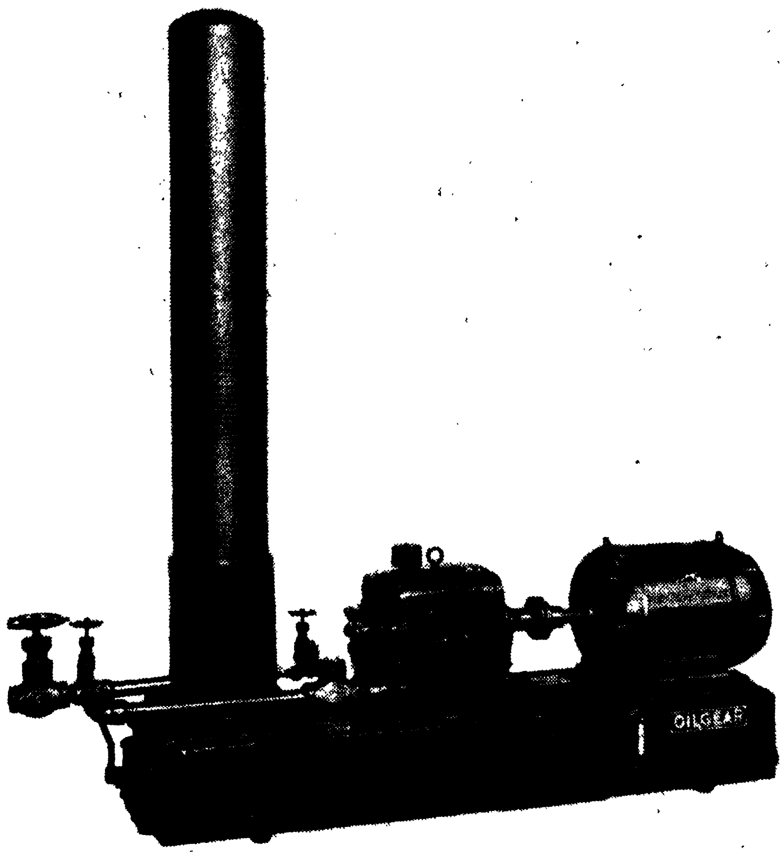


FIG. 105. Pump with hydraulic compensation control and pressure vessel. (*The Oilgear Co., Milwaukee, Wis.*)

rises, a tendency will be created to unbalance the control, causing it to go toward no-stroke position, which causes a reduction of delivery and pressure, again establishing equilibrium. A drop in line pressure has the opposite result. The spring will take charge and move the pump shift ring toward full-delivery stroke until equilibrium is again established. Great flexibility in cutoff differentials is possible with this control. For instance, if a control spring as above calculated is used, having a maximum force of 1,960 lb and a gradient of 588 lb for $\frac{3}{8}$ in., and assuming that we make the control piston 3 sq in., then the differential between full stroke and no stroke is 195 psi. This differential may be readily reduced

to almost any desired amount by increasing the control-piston area. The commercial models of the Vickers variable-delivery pump are equipped with this type of control.

A number of modifications of the pressure-compensating control are available. The control may be combined with a preset stroke adjustment by an auxiliary handwheel or setscrew, so that the pump may be set for any desired initial output. Figure 106 shows this type of control as made by Hydro-Power Inc. By using an auxiliary control plunger, three different pressure adjustments may be had by using either plunger independently or both in combination. This arrangement is useful in connection with double-acting cylinders, where full pressure is desired in the forward or working direction, and a lower pressure is satisfactory for

FIG. 106. Pressure-compensating control with auxiliary volume adjustment. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

the retraction or pullback direction. In that case, one plunger may be connected directly to the pump pressure line, and the other to the pullback connection of the cylinder, so that on the retraction stroke both plungers are effective.

In a good many applications, an initial reduction of pump discharge is desirable immediately before the hydraulic motor or cylinder operated by the pump meets resistance. This is particularly useful for plastic-molding presses, where the platens must come together rapidly until ready to engage, and then close slowly at a preset speed to build up the molding pressure. This control, developed by the author and built by Hydro-Power Inc., is shown in Fig. 107. In principle, it consists of the regular pressure-compensating control with the addition of a large auxiliary piston with an adjustable abutment to engage and reduce the stroke of the control crosshead. The amount of stroke may be preset by means of the adjustable abutment. The pressure required to raise the moving weight does not suffice to actuate the control. At an adjustable position in the travel of the press platen, a restriction is introduced in the pipe line, which

causes sufficient pressure to build up to actuate the stroke-reducing piston. The remainder of the travel is then made at reduced stroke until pressure builds up on the work, further reducing delivery to the zero-stroke position.

It should be kept in mind that on any of these controls, stroke reduction takes place until a minimum delivery is reached, where the pump delivers just enough oil to maintain the set pressure against leakage within itself and the hydraulic system. This affords an excellent means to check and compute the leakage losses in the system by measuring the pump stroke, if the pump is equipped with a stroke indicator, as all Oilgear pumps are.

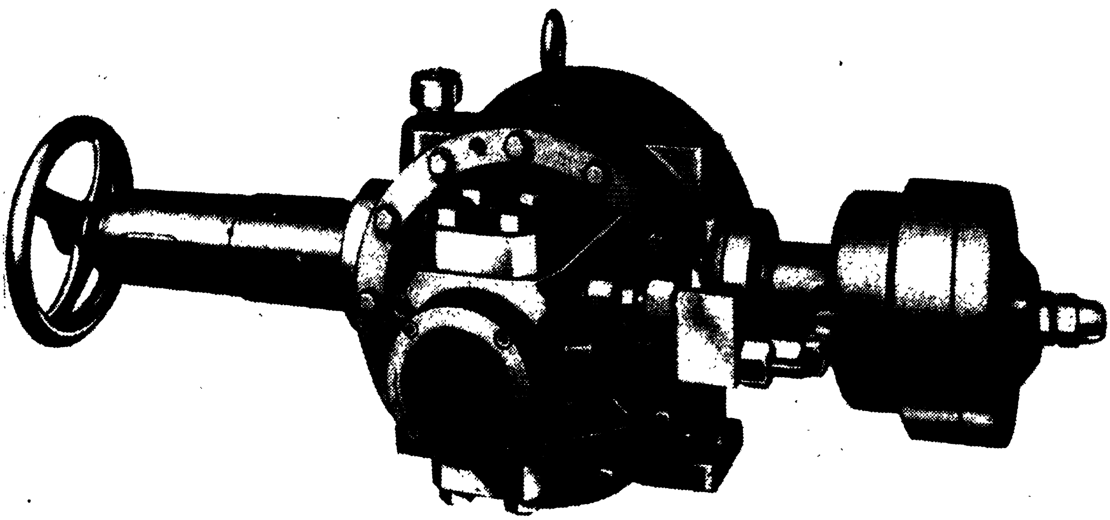


FIG. 107. Compensating and stroke-reducing control. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

For instance, if a pump is rated at $\frac{3}{8}$ in. eccentricity, or control stroke from neutral, and 20 gpm at full stroke, and if the stroke indicator shows $\frac{1}{32}$ in. control stroke from neutral, then the pump is delivering 1.665 gpm to make up leakage losses.

At times pressure-compensating controls, particularly those of the spring and plunger type, have a tendency to "hunt," or oscillate about the set pressure. This condition is aggravated by the presence of air in the system. Suction lines should be checked for tightness, and, if necessary, air bled from the cylinders and pipes. The tendency to hunt may generally be relieved by providing a small by-pass bleeding off sufficient oil to keep the pump from going past neutral and over on negative stroke, which invariably leads to serious oscillation. Careful workmanship and good finish is required on the moving parts of the controls to prevent sticking and binding with subsequent erratic behavior.

Servo-motor Control. So-called "servo-motor" controls are widely used for the control and actuation of reversing, or two-way, pumps.

Admittedly, the characteristics of reversing or two-way pumping systems are very desirable, owing to the absence of shocks on reversal, the smooth acceleration and deceleration of fluid flow, and the simplification of the hydraulic circuit by the elimination of four-way control valves. These advantages may be fully realized in a fast-acting machine, if rapid reversal of the pump-stroke control can be accomplished. We have seen that very heavy forces are involved in moving the pump-stroke control, and the power required to do this at rapid cycles would become quite excessive. Obviously, an auxiliary source of power, controllable so that the variable-delivery characteristics of the pump are retained and yet movement of the control crosshead may be accomplished with a minimum of effort, would be a most desirable solution.

Obviously, this purpose cannot be attained with a conventional oil-pressure cylinder and pilot valve. If such a device were used and the pilot valve shifted to admit pressure to one side of the piston, the pump crosshead would immediately shift in response to the pressure applied and would continue to shift until the pilot valve had been returned to its neutral position. Therefore, to obtain a desired delivery rate or crosshead position, the operator would have to jog the pilot valve in a series of short movements, to prevent the crosshead from overrunning the desired position. There would be no assurance that the pump would maintain its set delivery, as leakage through the pilot valve could cause the control piston to shift and alter the delivery stroke.

These shortcomings may be avoided by a tie-up between the pilot valve and power piston so that for any given increment of pilot-valve stroke, the resulting piston stroke will automatically reset the pilot valve to its neutral position and thus prevent further movement of the piston. Any inadvertant displacement of the piston immediately corrects itself by producing a corresponding pilot-valve movement, which will admit pressure fluid to restore the original position of the power piston.

The servo motor as such is quite old; the invention is attributed to one Farcot, a Frenchman who described it in a treatise "Le Servomoteur ou moteur asservi" in 1873. The first use of these devices was made in relay governors for steam turbines, and considerable literature was devoted to their theory and design as applied to the regulation of prime movers. In this text we shall confine ourselves to their application to the control of variable-delivery pumps, which has become a fertile field.

Figure 108 shows an elementary form of servo motor with restoring linkage. The clevis is attached to the pump shift ring, so that the stroke may be shifted by the movement of the power piston mounted in the servo-motor cylinder. A pilot valve is mounted above the power piston and so arranged that the two valve heads exactly cover two annular slots

in the valve body, which communicate through suitable passages with both sides of the power piston. Pressure from an auxiliary pump is supplied at the center of the pilot valve between the two valve heads. An exhaust opening is provided, as indicated, permitting exhaust of pressure from either end of the pilot valve through a bore in the pilot-valve plunger.

The valve may be operated by means of a hand lever that is mounted on a fulcrum that is not fixed, but travels with the power-piston rod. If the hand lever is now swung in clockwise direction, the pilot valve will move to the right, admitting pressure to the right-hand side of the power piston, causing it to travel to the left. Momentarily, the hand of the operator holding the valve lever becomes the fixed fulcrum about which

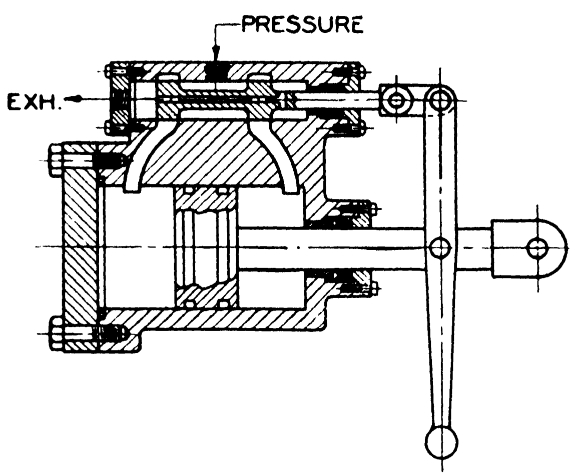


FIG. 108. Servo motor with restoring linkage.

the lever will turn, being pulled to the left by the power piston. This movement restores the pilot valve to its original neutral position. The final result is that the hand lever assumes a position corresponding to that of a fixed fulcrum at the top, and the action has taken place as if the operator had moved the power piston manually by means of the hand lever turning about its upper fulcrum. Thus the operator in effect moves a heavy load with little effort a controlled distance, determined by the desired pilot-valve stroke. A similar action will take place in the opposite direction. In case of accidental displacement of the power piston, the valve will be displaced and cause the position of the piston to be restored, provided that the lower end of the lever is either held in position or locked in place by mechanical means.

A modern version of this old device is illustrated in Fig. 109. In this design all external linkage is eliminated, and the power piston follows the servo valve in proportion to the servo-valve displacement. The pressure enters at the connection indicated and acts permanently against the

differential area of the power piston, tending to move it and the pump shift ring connected to it to the left. Pressure may enter the radial holes in the undercut position of the power piston and pass into an annular groove that is closed by a "line-on-line" cutoff by the pilot valve. Similar passages closed off by the opposite end of the pilot valve lead to the large area of the power piston and may connect it to the interior of the pilot-valve bore and from there into the exhaust. A slight movement of the pilot valve to the right will cause admission of pressure to the head end of the power piston, which in turn will overcome resistance of the pressure on the smaller differential area and move the power piston to the right, until the valve opening is again closed and the device comes to rest. Conversely, if the valve is moved to the left, oil is exhausted from the

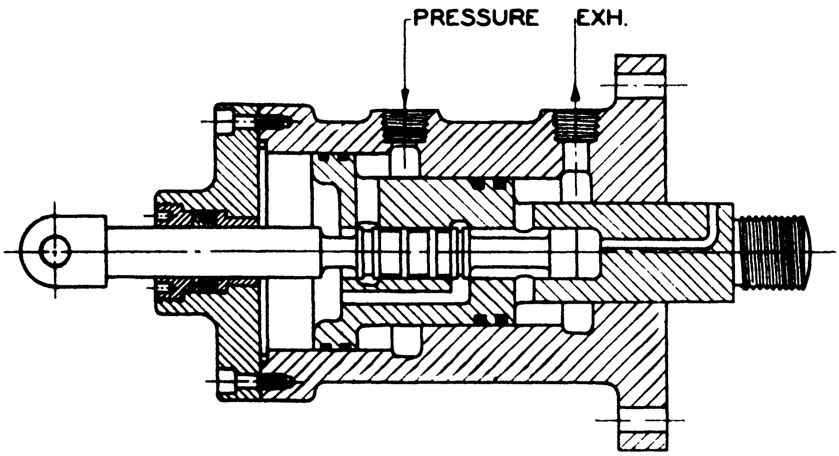


FIG. 109. Full hydraulic servo motor.

head end, and the piston will move to the left under the influence of the constant pressure in the differential area. Care must be taken in the design of these devices to have the valves well balanced to prevent sticking and locking. To this end, the valve is grooved circumferentially, all but relatively narrow ledges being cut away to prevent pressure fluid from squeezing between valve and valve bore and locking the valve. The exhaust end of the valve also carries a balancing piston to prevent back pressure from the exhaust from interfering with the action of the valve.

In designing a servo motor, care must be taken to make passages and openings of sufficient size to carry the flow of oil for which the device was designed. Hydraulic working pressures of from 150 to 500 psi are used in these devices, generated by small auxiliary pumps of the gear or vane type. Sufficient capacity should be available to move the shift ring from "hard over" to "hard over" in from 0.3 to 0.5 sec.

A few figures made for the 20-gpm unit designed previously will illustrate this. The total control force F was 2,050 lb. With a hydraulic pilot pressure of 250 psi, the

differential area should be $^{2050}/_{250} = 8.2$ in. The head should be twice this area or 16.4 sq in. To traverse crosshead ¾ in. in 0.3 sec will require

$$\frac{8.20 \times 0.75 \times 60}{0.3 \times 231} = 5.35 \text{ gpm}$$

A ½-in. valve and passages are satisfactory.

Figure 110 shows a very unusual servo control manufactured by the Oilgear Co. Pressure is continuously maintained on the small piston,

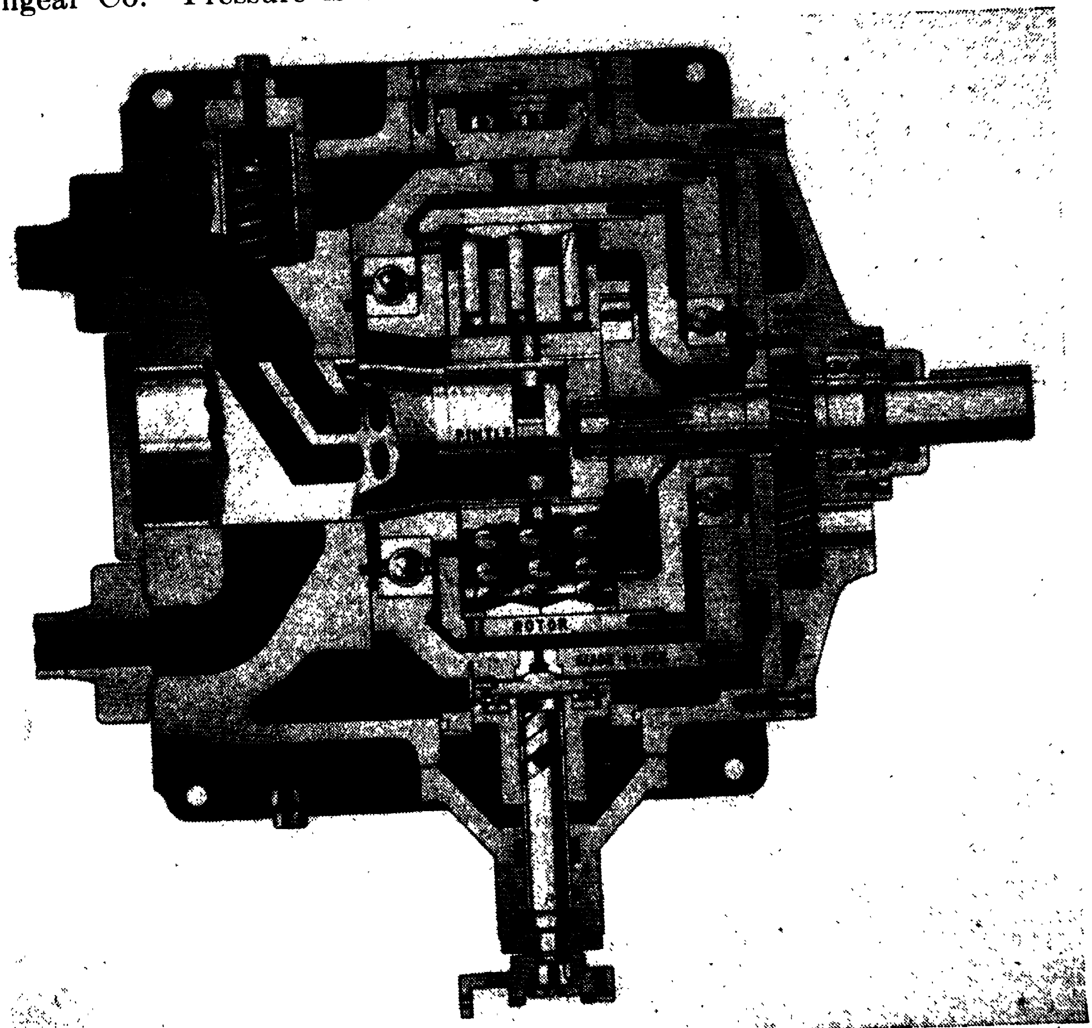


FIG. 110. Pump with servo control. (*The Oilgear Co., Milwaukee, Wis.*)

while it may be selectively supplied or exhausted through spiral slots in the servo valve, which are brought successively into engagement with an opening in the valve body. The great care used in the Oilgear design is well illustrated in the provision of the small ball bearing between servo-valve-and-piston assembly and the pump shift block, which permits the shift block to "breathe" up and down between the clearances in the shift pads without cramping or binding the servo valve. An exterior view of the same device is shown in Fig. 111.

Servo controls exhibit at times a tendency to "hunt." This is generally due to excessive valve speed (power piston unable to catch up), loose or worn linkage, or air in the hydraulic system.

Precision Controls. The Oilgear Company in Milwaukee manufactures a precision control that rigorously maintains constant output of a variable-delivery pump despite fluctuation in load, temperature, and power supply. For the description of this control, the following is quoted from the manufacturers' bulletin with their permission:

Disk-type DT Precision Speed Control (Differential-time). The DT control unit is the heart of this system, which converts the fluid power

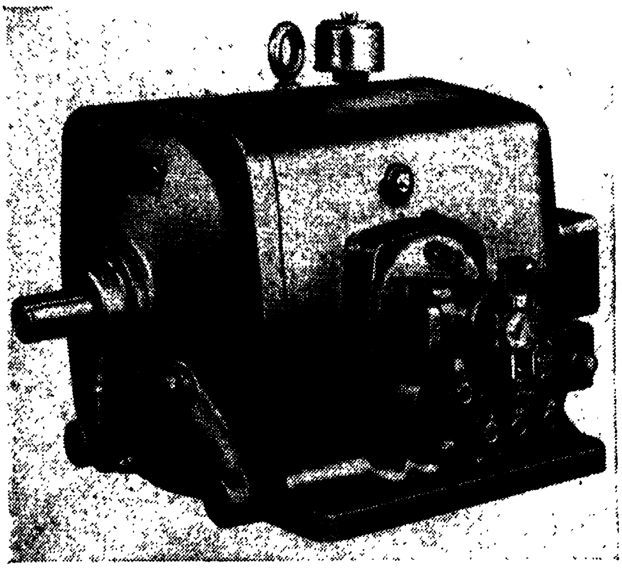


FIG. 111. Pump with servo control, exterior view. (*The Oilgear Co., Milwaukee, Wis.*)

transmission from merely an adjustable-speed drive to the most precise and flexible industrial driving means now obtainable.

It is well known that hydraulic power transmissions heretofore used will depart somewhat from their set speeds as changes occur in load and temperature. While the amount of departure is well within the limits permitted, on many commercial applications, the requirements of other machines and processes are such as to demand precise and uniform speed control.

The departure from set speed in hydraulic transmissions is caused by the leakage of oil past the running fits. The hydraulic motor tends to run more slowly when the load increases, when the temperature of the oil increases and the viscosity of the oil is reduced, or if the transmission wears, permitting more oil to escape past the running fits.

The essential principle of the DT precision control is that the hydraulic motor, driven by the flow of power from its variable-stroke pump, simultaneously transmits the resulting speed back to a differential com-

parator. This differential also receives a standard or master speed from a small synchronous motor (time control), or from some other unit, roll stand, or float roll (measuring control). In any case, the differential continuously compares the actual speed with the master speed and translates every discrepancy into exactly the increase or decrease of pump discharge necessary to correct the error. The action of the disk-type control is so quick that fluctuations of speed due to ordinary load changes are caught and corrected within about 1/10 sec. With moderate load

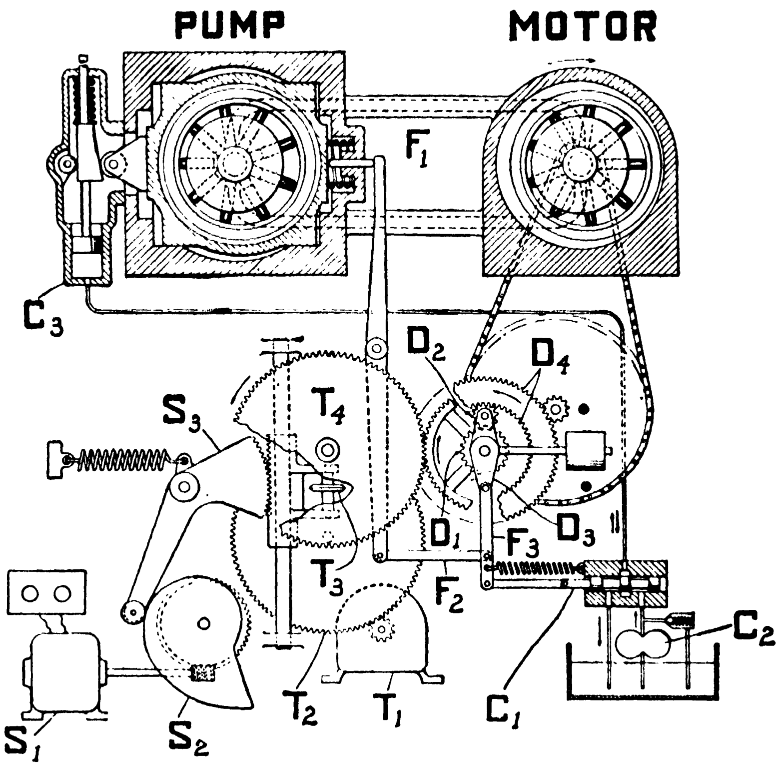


FIG. 112. Disk-type precision speed control. (*The Oilgear Co., Milwaukee, Wis.*)

fluctuations, such as usually exist in a continuous processing plant, the momentary governing errors are held within 1/4 of 1 per cent plus or minus, while the integrated error over a period of time (say 10 sec or more) is not measurable by a stop watch.

The DT disk-type control is shown in its usual form in Fig. 112 with a small synchronous or Selsyn motor T_1 furnishing the master speed. If T_1 is a synchronous motor, the control is by time, and the speed of the hydraulic motor is in rpm. If T_1 is a Selsyn motor, it receives its current from a Selsyn generator driven by its master unit, and follows the speed of that master through its entire range from a standstill to maximum.

The master speed is transmitted to the differential D through the friction disk-ratio changer, comprising a movable idler disk T_3, spring-pressed between the driving (master) disk T_2 and the driven disk T_4.

Disk T_3 is mounted in a slide block, adjustable by a push-button control motor S_1 (or by manual or float-roll control) through cam S_2 and segment S_3.

The master speed as modified by the ratio changer, is transmitted by gear teeth in edge of T_4 to the sun pinion D_1, the first leg of the differential comparator already mentioned. Planet gear D_2 is mounted on a radial valve-actuating crank D_3 (third leg of differential), which actuates pilot valve C_1 through floating lever F_3. Planet gear also meshes with internal ring gear D_4, exactly twice the sun-gear diameter. Any movement of pilot valve resulting from movement of axis of planet gear and crank D_3 changes the pump stroke and hydraulic-motor speed. To hold pilot valve stationary, ring gear D_4 (second leg of differential) must be driven by hydraulic motor M at exactly one-half the standard speed of gear D_1. Any discrepancy will result in a slight movement of valve C_1, permitting oil from gear pump C_2 to enter pump-stroke-changing cylinder C_3 and increase pump discharge, or permitting oil to escape from C_3 and reduce pump discharge. When the change in pump discharge has corrected motor speed to balance the speeds of the differential gears again, the pilot valve will resume its neutral position.

In order to prevent hunting, an essential part of the mechanism is the follow-up lever F_1, which responds to the movement of pump slide block and by means of link F_2 closes the pilot valve as soon as the required correction has been made.

As the disks T_1, T_2, T_3 are made of hardened steel, accurately ground, and transmit no appreciable power, they show no wear after long service. This is true even when the speed is not changed and idler disk T_2 runs in one location continuously.

CHAPTER VIII

THE UTILIZATION OF OIL HYDRAULIC POWER

1. Rotary Hydraulic Motors. In Chap. VII we have seen how hydraulic power is generated by means of rotary pumps, supplied with power by prime movers such as electric motors. These pumps supply oil, flowing under pressure in conduits, to carry the hydraulic power to the point of application. There the fluid power may again be converted into mechanical power expressed in torque and rotating speed. This is done by rotary hydraulic motors.

Almost all designs of hydraulic pumps may, at least theoretically, be used as hydraulic motors, if hydraulic power in the form of oil flowing under pressure is supplied to them. Actually there are only a few commercially successful designs of motors in use, for a number of reasons.

First, only a few of the different designs of rotary pumps will make efficient motors, for inherent hydraulic and mechanical reasons. Second, rotary hydraulic power has been and is relatively little used, owing to the abundance of economically priced, readily available electrical motors. Only where special considerations dictate their use have hydraulic transmissions and rotary motors gained a foothold and displaced electrical power for producing rotary motion and torque. This refers in particular to cases where extreme range of speed adjustment is required, which cannot be met by electrical units, or where compactness is a prime requisite not obtainable by any other means. Hydraulic motors have been developed that have more horsepower in less space and weight than any other known source of power. In cases where hydraulic power is available, as on a hydraulically operated machine tool, press, or other machine, the designer may do well to give consideration to the use of hydraulic motors for auxiliary functions. Hydraulic transmissions have found an excellent field of application in paper-machine and printing-press drives and in other fields where extreme range and close adjustment of speeds are required.

Rotary motors now in commercial production are either of the gear or plunger type. The most important representative in the former category is the Vickers hydraulic gear motor. We have shown in Chap. VII that heavy unbalanced loads exist in gear pumps. The Vickers gear motor, illustrated in Fig. 51, avoids this objectional feature. This is done by

providing a pocket approximately diametrically opposite the pressure port and conducting pressure from the pressure port to this pocket.

Another method, used successfully by the author, is to drill holes from one tooth space diametrically through the gear into the diametrically opposite tooth space. (Gears must have an even number of teeth and must be made integrally with their shafts.) There are as many drills as there are teeth or tooth spaces, and the drills must, of course, not interfere with each other. The result will be almost perfect balance and excellent operation as gear motors at pressures as high as 1,000 psi.

Torque, Horsepower, and Efficiency. The primary purpose of a hydraulic motor, like that of any motor, is to develop torque. Hydraulic pressure supplied to the motor acts on the surface of the teeth, pistons, or vanes and creates a force, and this force or a component of it, acting tangentially to the distance from the center of rotation, produces a turning moment or torque. Torque may be expressed in terms of horsepower and speed as follows.

A force F acting at a distance from the center of rotation, L, produces a moment or torque F times L. The work done by this force is $2\pi FL$. If F times L is denoted as T, then the work the power developed is $2\pi Tn$ ft-lb per min, or expressed in horsepower:

$$HP = \frac{2\pi Tn}{33{,}000} \tag{1}$$

$$= \frac{Tn}{5{,}250} \tag{2}$$

where T = torque, ft-lb

Torque may be related to pressure and displacement in the following manner. If the displacement of the pump in cubic inches per revolution is denoted as q, we have

$$HP = \frac{qnp}{33{,}000 \times 12} \tag{3}$$

Equating (1) and (3), we have

$$T = \frac{qp}{24\pi} \qquad \text{ft-lb} \tag{4}$$

with p in pounds per square inch. Equation (4) gives the average theoretical torque for any design of hydraulic motor and is independent of the geometrical configuration.

Efficiencies of hydraulic motors may be expressed in a manner similar to those of hydraulic pumps. The volumetric efficiency of a motor is the ratio between the geometric displacement at a given speed and gross input as measured at a given pressure and the same speed.

$$e_v = \frac{Q_g}{Q_i} \times 100 \qquad \text{in per cent} \tag{5}$$

The mechanical efficiency is the ratio between the actual output horsepower, as determined by measuring output torque and speed, and the hydraulic horsepower of the *geometric* displacement at that speed.

$$e_m = \frac{HP_a}{HP_h} \times 100 \qquad \text{in per cent} \tag{6}$$

Total over-all efficiency is the product of volumetric and mechanical efficiency, or

$$e_t = e_v e_m \tag{7}$$

or

$$e_t = \frac{Q_g}{Q_i}\frac{HP_a}{HP_h} \tag{8}$$

Since $HP_h = 0.000583Q_g p$,

$$e_t = \frac{HP_a}{Q_i \times 0.000583p} \tag{9}$$

Example: In the following an analysis will be made of the gear pump calculated in Chap. VII to determine its characteristics as a motor. The pump has gears of 5 DP, a 2.6 pitch diameter, and 1.3 in. width. Outside diameter is 3 in., and working depth diameter, 2.2 in.

$$q = \left(\frac{\pi d_0^2}{4} - \frac{\pi d_i^2}{4}\right) w$$

$$= (7.05 - 3.80)1.3 = 4.26 \text{ cu in. per rev}$$

$$\text{Torque } T = \frac{4.26 \times 1{,}000}{24\pi} = 56.5 \text{ ft-lb}$$

At 1,200 rpm the hydraulic horsepower may be computed as follows:
From torque:

$$HP_h = \frac{56.5 \times 1{,}200}{5{,}250} = 12.9$$

From displacement and pressure:

$$HP_h = \frac{4.26 \times 1{,}200 \times 1{,}000}{33{,}000 \times 12} = 12.9$$

If the mechanical efficiency is assumed to be 85 per cent, then

$$HP_a = 12.9 \times 0.85 = 10.95$$

$$\text{Actual torque} = \frac{5{,}250 \times 10.95}{1{,}200} = 48 \text{ ft-lb}$$

$$Q_g = \frac{4.26 \times 1{,}200}{231} = 22.2 \text{ gpm}$$

$$e_v = 90 \text{ per cent} \qquad Q_i = \frac{22.2}{0.9} = 24.6 \text{ gpm}$$

$$e_t = \frac{10.95 \times 100}{24.6 \times 0.000583 \times 1{,}000} = 76.5 \text{ per cent}$$

A comparison of hydraulic motor characteristics with that of a corresponding pump is interesting. On the pump previously calculated, the input horsepower was 15.3, the output horsepower was $15.3 \times 0.765 = 11.7$. The motor output horsepower is 10.95, and the input horsepower, 14.3. This indicates that although both units have identical physical dimensions and the same efficiencies, the pump is capable of absorbing more horsepower than the motor. This apparent discrepancy is due to inherent characteristics of hydraulic drives and may be explained as follows. A hydraulic unit, pump or motor, is limited in its power capacity by its mechanical strength, which limits its maximum operating pressure, and by its maximum speed, limited by bearing design or similar considerations. The ability of a hydraulic pump to absorb power, therefore, depends on its volumetric efficiency, provided that ample power is available to overcome mechanical losses, while the power capacity of a motor depends entirely on its mechanical efficiency, assuming that sufficient volume is available to run it at its maximum rated speed, regardless of volumetric losses. If the volumetric efficiency of a unit is higher, therefore, as in this case, it makes a better showing as a pump, while with higher mechanical efficiency more power can be developed as a motor. If mechanical and volumetric efficiencies are the same, then the power absorption will be the same, whether it is operated as pump or motor.

Radial- and Horizontal-plunger-type Rotary Motors. Generally developed Eqs. (4), (5), (6), and (9) are applicable to rotary plunger-type motors. These motors are adaptations of the pumps described in Sec. 4, Chap. VII. Not every design of rotary pump, however, makes a good motor owing to inherent design characteristics that will be dealt with in the following. Commercially available motors at present are the Oilgear, Vickers, Waterbury, and Sundstrand. All these units employ kinematics with finite connecting-rod length or the equivalent thereof.

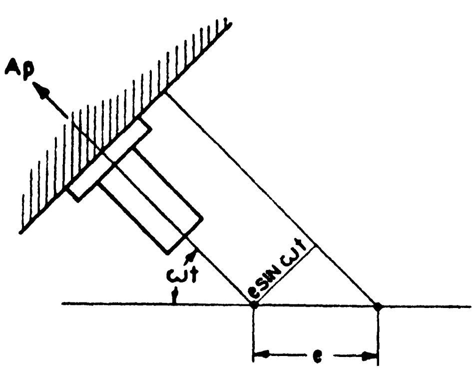


FIG. 113. Analysis of torque (infinitely long connecting rod).

ANALYSIS OF A MOTOR WITH INFINITELY LONG CONNECTING ROD. A mechanism of this kind is shown in Fig. 113. Hydraulic pressure p acting on the area A of the piston produces a force Ap. It becomes apparent that this force cannot produce a torque or turning moment on the primary rotor. Therefore torque must be created on the secondary rotor. This torque has the magnitude

$$T = pAe \sin \omega t \qquad \text{in.-lb} \tag{10}$$

It represents the instantaneous torque of one piston. If it is desired to drive the primary rotor, torque must be transmitted from the secondary to the primary through the medium of the pistons. This retransmission of torque results in additional losses, which reduce the efficiency of the unit. This is the reason why no motors with this type of drive have made their appearance.

From Eq. (10) we may compute the average torque of one piston for one complete cycle as follows:

$$T_{\text{avg}} = \int_0^\pi \frac{T\,d(\omega t)}{2\pi} = \frac{epA}{2\pi}\int_0^\pi \sin \omega t\,d(\omega t) = \frac{epA}{\pi} \tag{11}$$

Equation (11) will revert to Eq. (4) with $q = 2eA$ and T expressed in foot-pounds.

By comparing Eq. (10) with Eq. (26), Chap. VII, we find the expression for torque and displacement identical in construction. The entire analysis that has been given for the displacement variation in multiple-plunger pumps, therefore, applies to torque variations. Reasonably uniform torque may be expected in multiple-plunger motors having an odd number of pistons, and output speed variations correspond to variations in torque, speed being proportional to torque.

Mention should be made of the fact that a motor of very good efficiency may be designed on the principle illustrated in Fig. 113, if power is taken off the secondary rotor. In that case, no driving torque is transmitted through the pistons; as a matter of fact, pistons are not subjected to any lateral forces other than those created by friction of the crossheads on their guides and friction of the rotor on its bearings. Some design difficulties may be encountered in driving off the secondary rotor, particularly in case of variable-delivery motors.

The analysis of a motor with finite length of connecting rod may be made easily by analogy. We have seen that the instantaneous output of one piston is

$$Q_I = A\omega e\left(\frac{e}{R}\sin \omega t \cos \omega t + \sin \omega t\right) \qquad \text{[Eq. (38), Chap. VII]}$$

Instantaneous input to one piston is

$$Q_I = A\omega e\left(\sin \omega t - \frac{e}{R}\sin \omega t \cos \omega t\right)$$

From this follows the instantaneous torque analogous to Eq. (10) and Eq. (26), Chap. VII:

$$T = Ape\left(\sin \omega t - \frac{e}{R} \sin \omega t \cos \omega t\right) \tag{12}$$

A computation of the average torque leads to the same value as that expressed in Eq. (11).

By referring to Fig. 114, it may be seen that in a motor with finite connecting-rod length, torque is directly created on the primary rotor by a component of the piston thrust. The Oilgear hydraulic motor operates in this manner. The moment is FP. Expressing F and P in terms of e, R, and ωt results in Eq. (12).

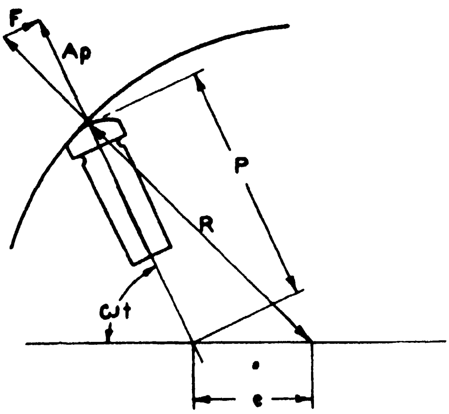


FIG. 114. Analysis of torque (finite connecting rod).

Design of Radial-plunger-type Motors. Design of these motors follows closely that of radial-type pumps. As pointed out, only pumps with finite-connecting-rod kinematics will operate efficiently as motors, if torque is delivered from the primary rotor.

Motors may be made in both constant- and variable-displacement models. Constant-displacement motors are constant-torque machines. Horsepower varies as speed, and the torque is the same at all speeds, being determined by hydraulic pressure. Variable-displacement motors are variable-torque and constant-horsepower machines, and torque varies inversely as speed. The speed range for constant-horsepower, variable-speed motors generally does not exceed 3:1, owing to speed limitations of commercial units.

Oilgear constant-displacement motors correspond in design to their constant-displacement pump shown in Fig. 87. Motors are available in eight sizes, ranging from 135 to 8,125 in.-lb normal torque. Working pressures at normal torque are approximately 1,200 and 1,800 psi. Permissible peak torques are 150 per cent of normal torques. Maximum operating speeds are from 1,090 to 800 rpm, and minimum speeds are 5 to 30 rpm. This results in output horsepowers at maximum speed of from 2 to 100, filling a wide range of requirements. Torque efficiencies of the Oilgear motor are excellent, running practically constant at 95 per cent for the range of 1,200 down to 100 rpm on a 20-hp unit, to drop to 80 per cent stalled torque.

Variable-displacement motors are available in a similar range of torques and operate at lower full-torque speeds to obtain a 3:1 speed range without exceeding maximum permissible speeds. These range from 2,450 rpm for the smallest to 1,380 rpm for the largest unit.

Combinations of pumps and motors may be made to form transmissions to develop almost any desired characteristics. Combination of variable-delivery pumps with constant-displacement motors results in constant-torque transmissions. If geometric displacement of both units is the same, rotating speeds of motors are somewhat slower than those of the pumps owing to leakage in both pumps and motors. A transmission of the type described is not a speed-reduction unit in the accepted sense, but rather a variable-speed drive.

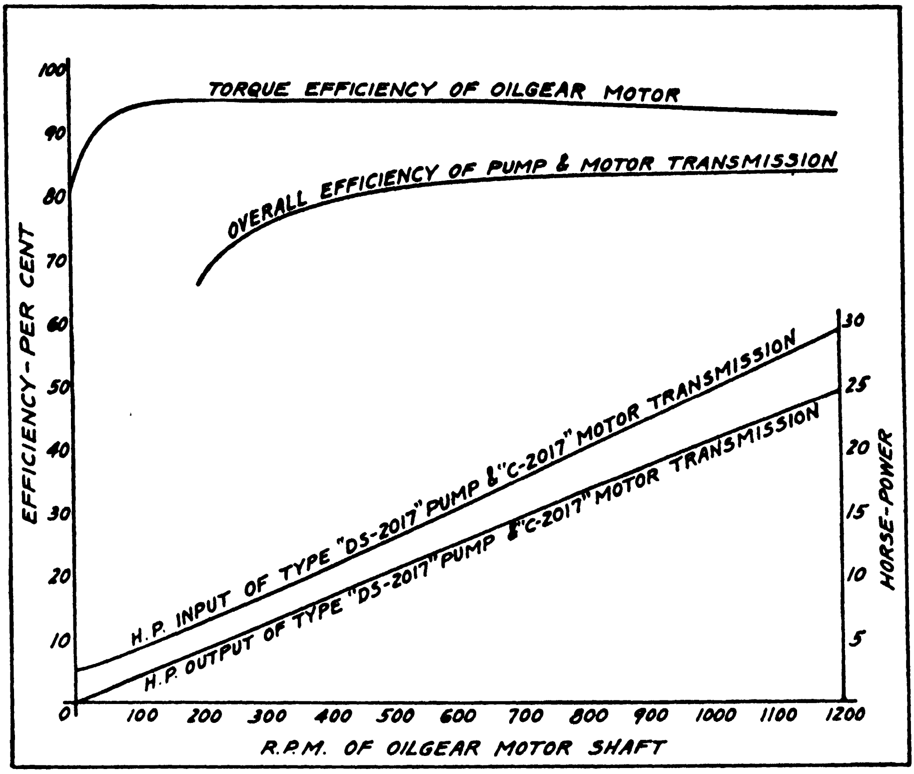


FIG. 115. Performance curves at full load rating of separate pump and motor transmission 4 ft. or less apart. Operating temperature: 120°F. Oil viscosity: 300 SSU at 100°F. (*The Oilgear Co., Milwaukee, Wis.*)

Combination of constant-delivery pumps and variable-delivery motors results in a constant-horsepower, variable-torque transmission. Horsepower is constant over the permissible speed range of the motor, generally 3:1. In constant-horsepower Oilgear units, constant-delivery pumps of a given size are combined with the variable-delivery motor of the next larger size, resulting in about one-half the normal full-torque motor speed. Reduction in stroke to one-third of full stroke produces a maximum speed of $1\frac{1}{2}$ times the normal speed.

Efficiency curve of an Oilgear separate pump and motor transmission is shown in Fig. 115. In the design of these transmissions, pipe-line and port velocities should be held within 5 to 10 ft per sec. Suitable make-up check valves must be provided to make up leakage losses in the units.

Oilgear units have supercharging pumps to keep the transmission under a slight head of pressure at all times. Figure 116 shows pumping unit with make-up checks and relief valve.

Oil-tank capacities providing for approximately 1 min of output of the pump have proved satisfactory for separate pump and motor transmission. For best efficiency, radial-type plunger motors should develop

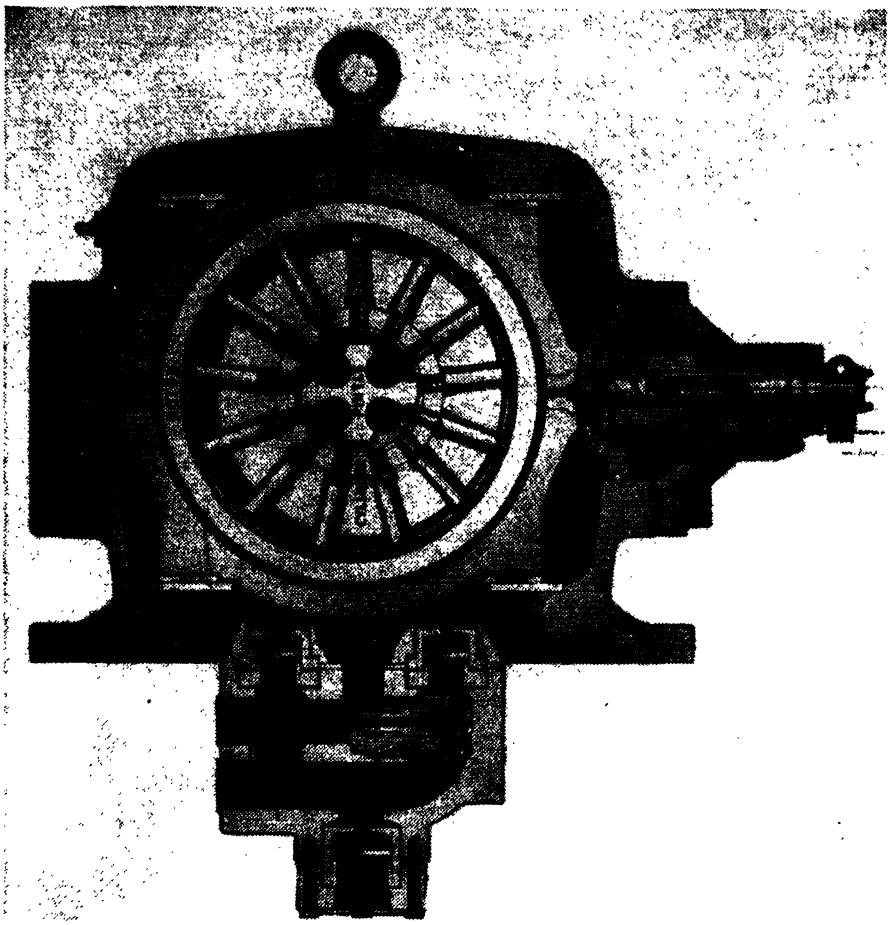


FIG. 116. Oilgear pumping unit with make-up check and relief valves. (*The Oilgear Co., Milwaukee, Wis.*)

their normal torque at about 1,500 to 1,750 psi. Oil of about 300 to 350 SSU viscosity has been found most satisfactory. For severe service and continuous operation at maximum loads, water cooling of the oil may be needed. On Oilgear transmissions the supercharging pump may be utilized for this purpose by passing its discharge through an oil cooler.

Horizontal-plunger-type Motors. Design of these motors follows closely that of the horizontal-plunger-type pumps described in the preceding chapter, with the exception of modifications in the porting arrangement and the fact that the swash-plate angle is fixed.

Example: Design of a hydraulic motor may be illustrated by calculating a companion motor to go with the pumping unit discussed in the preceding chapter. The net displacement of the pump was 43 gpm. In order to operate the motor at approximately the same speed as the pump at an intermediate load and to compensate for leakage losses, we make the piston diameter slightly smaller.

Piston diameter = $1\frac{7}{32}$ in. Stroke = $1\frac{3}{8}$ in.
Maximum swash-plate angle = 20°

Radius of piston circle will then be approximately

$$R = \frac{e}{\tan 20^\circ} = \frac{0.6875}{0.364} = 1.875$$

This is an approximate value for R, neglecting the connecting-rod tilt. Generally the mechanism should be so designed that the connecting rod is horizontal at one-half the swash-plate angle. To this end, the cylinder barrel is laid out with R as computed, and rods are made horizontal at one-half the swash-plate angle. Rods will then tilt toward or away from the shaft center when swash-plate angle is increased or decreased. Slight corrections may then be made on angle, eccentricity, or cylinder center radius, whichever is most convenient.

Volumetric and mechanical efficiencies are assumed to be 95 per cent each. With the motor supplied with a volume $Q_i = 43$ gpm from the pump, Q_g must be $Q_i e_v = 41$ gpm. With

$$Q_g = 2AenN/231, \text{ we obtain}$$

$$n = \frac{41 \times 231}{2 \times 0.6875 \times 1.16 \times 7} = 850 \text{ rpm}$$

With $e_m = 0.95$,

$$HP_a = 0.95 HP_h$$
$$HP_h = Q_g \times 0.000583p = 24 \text{ hp}$$
$$HP_a = 0.95 \times 24 = 22.75 \text{ hp}$$
$$e_t = \frac{Q_g}{Q_i}\frac{HP_a}{HP_h} = \frac{41}{43}\frac{22.75}{24} = 0.90$$
$$\text{Output torque} = \frac{5{,}250 \times 22.75}{850} = 140 \text{ ft-lb}$$
$$\text{Efficiency of pump and motor combination} = \frac{22.75}{28} = 0.81$$

Valve-plate Calculation for Motor. To compute the land area on the port plate, we use Eq. (41), Chap. VII, with an average pressure-ratio factor of $m = 1.45$.

$$AN = 1.16 \times 7 = 8.1$$

Therefore

$$\pi(d^2 - a^2) = 8.1 \times 1.45 - 5.15 \times 0.45 = 11.8 - 2.3 = 9.5$$
$$d = 2\tfrac{9}{32}$$
$$a = 1\tfrac{1}{2}$$
$$\frac{b - a}{b - c} = \frac{5}{6} = 0.83$$

In the preceding calculations, the pitch diameters and spacings in cylinder bores of both motor and pump were made the same for all

pistons in both units. If this is done, irregularities will be introduced in displacement and output speed, owing to the universal-joint error introduced by the swash-plate drive. These irregularities may be compensated for by offsetting the cylinder spacing in pump and motor. The motor is compensated for the 20° angle by changing both the circumferential and radial spacing of the cylinder bores, while the pump is compensated for an intermediate angle with only the angular spacing changed. In the motor some of the cylinder bores are nearer the center and others farther than the pump-cylinder bores. The compensation for the angular-velocity variations is based on the following analysis.

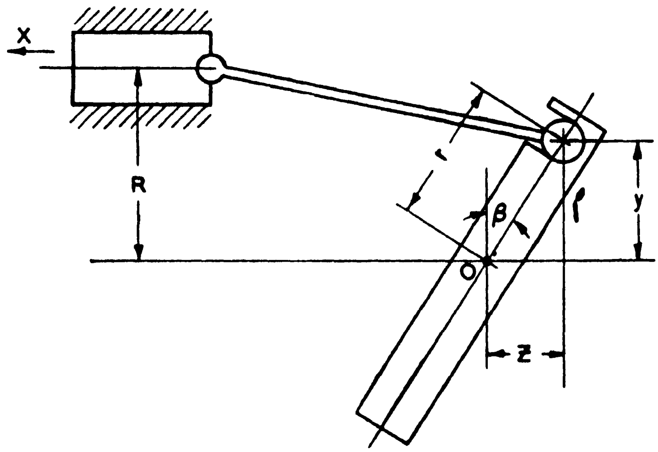


FIG. 117. Horizontal-piston-drive mechanism.

In Fig. 117 the mechanism is shown in one of its dead-center positions. Starting with the piston fully retracted toward the right, the distance the piston moves while the shaft rotates an angle ωt is given by

$$x = r \sin \beta - \sqrt{r^2 - y^2} - \sqrt{L^2 - (r \cos \beta - R)^2} + \sqrt{L^2 - (y - R)^2} \tag{13}$$

This relationship is correct only if the shaft center line, piston center line, and connecting-rod socket remain in one plane. This holds true for the one piston that is located on the axis of the universal-joint trunnion located in the plane of the swash plate. The third and fourth members in Eq. (13) represent the correction due to the connecting-rod tilt. In order to simplify the analysis, we will neglect this connecting-rod tilt and assume that the connecting rod is infinitely long.

To determine y as function of r, β, and ωt, we proceed as follows: The equation of the ellipse formed by projection of the socket circle on the swash plate upon a plane perpendicular to the shaft axis is, in polar coordinates,

$$y^2 = \frac{r^2 b^2}{b^2 \sin^2 \omega t + r^2 \cos^2 \omega t} \tag{14}$$

where r = major radius
b = minor radius = $r \cos \beta$
Hence

$$y^2 = \frac{r^4 \cos^2 \beta}{r^2 \cos^2 \beta \sin^2 \omega t + r^2 \cos^2 \omega t} \tag{15}$$

From this we obtain by a few trigonometric manipulations

$$y^2 = \frac{r^2}{1 + \cos^2 \omega t \tan^2 \beta} \tag{16}$$

Combining Eqs. (13) and (16), we have

$$x = r\left(\sin \beta - \sqrt{\frac{\cos^2 \omega t \tan^2 \beta}{1 + \cos^2 \omega t \tan^2 \beta}}\right) \tag{17}$$

If we stipulate the angle ϕ such that

$$\tan \phi = \cos \omega t \tan \beta \tag{18}$$

then

$$\sin \phi = \sqrt{\frac{\tan^2 \phi}{1 + \tan^2 \phi}} \tag{19}$$

and

$$x = r(\sin \beta - \sin \phi) \tag{20}$$

ϕ is the angle between a line connecting swash plate and rod-socket centers and a plane perpendicular to the shaft.

Equation (20) shows that the displacement of the piston located on the axis of the secondary universal-joint trunnion axis is sinusoidal with respect to the shaft angle ωt. This is not true for any other piston axis, owing to the universal-joint error. If the angular location of the connecting-rod socket in the plane of the swash plate relative to the above "reference" piston is denoted by ξ, and if the corresponding angle in a plane perpendicular to the shaft is denoted by g, we may compute g as follows: We have

$$\tan \varphi = \tan \omega t \cos \beta \tag{21}$$

where φ is the angular travel of the swash plate corresponding to the angular travel ωt of the shaft.[1] Then

$$\tan (\varphi + \xi) = \tan (\omega t + g) \cos \beta \tag{22}$$

[1] This is the universal-joint formula. Compare R. C. H. Heck, "Mechanics of Machinery," 1st ed., Vol. 2, "Mechanism," p. 340, McGraw-Hill Book Company, Inc., New York, 1923.

By eliminating φ and solving for g, we obtain

$$\tan g = \tan \xi \frac{a + b \cos 2\omega t}{c + d \sin 2\omega t} \tag{23}$$

where $a = 2 - \sin^2/\beta$
$b = \sin^2/\beta$
$c = 2 \cos \beta$
$d = \tan \xi \sin^2/\beta$

The angular universal-joint error is then $g - \xi$, whence the circumferential error is $y\ (g - \xi)$. From this we compute

$$\Delta x = y(g - \xi) \tan \gamma \tag{24}$$

where

$$\tan \gamma = \frac{dz}{d(y\omega t)} \tag{25}$$

Since y varies very little (between $r \cos \beta$ and r), we may substitute a constant value y_{avg} and write

$$\tan \gamma = \frac{dz}{y_{\text{avg}}\, d\omega t} \tag{26}$$

From Fig. 117 we have

$$z^2 = r^2 - y^2 \tag{27}$$

Also

$$y^2 = \frac{r^2}{1 + \cos^2 \omega t \tan^2 \beta} \tag{16}$$

Combining (27) and (16), we have

$$z = y \cos \omega t \tan \beta \tag{28}$$

Again substituting y_{avg} for y, we obtain

$$\tan \gamma = -\tan \beta \sin \omega t \tag{29}$$

From this

$$\Delta x = -y_{\text{avg}}(g - \xi) \tan \beta \sin \omega t \tag{30}$$

With the aid of Eqs. (23) and (30) we may compute the correction Δx for any connecting-rod-socket location ξ (relative to the reference piston) on the swash plate at any given shaft angle ωt. Adding this to x from Eq. (20) for the same shaft angle ωt, we obtain the actual displacement of the particular piston as function of the shaft angle.

These displacements may be plotted, and from the plot the piston speed or torque relationship may obtained by graphic differentiation. Resultant plots of the combined output speeds or torques may be corrected by changing the phase angles or amplitudes on the graph.

It should be pointed out that it is possible to express both x and Δx as functions containing fundamental and first harmonics of ωt. These functions may be differentiated, and an analytical expression obtained for the speed or torque values as functions of ωt. An analytical expression may also be set up of the resultant ripples from the combination of the piston speeds or torques, which may then be corrected in the manner indicated to obtain smoother delivery or torque. This analysis is quite

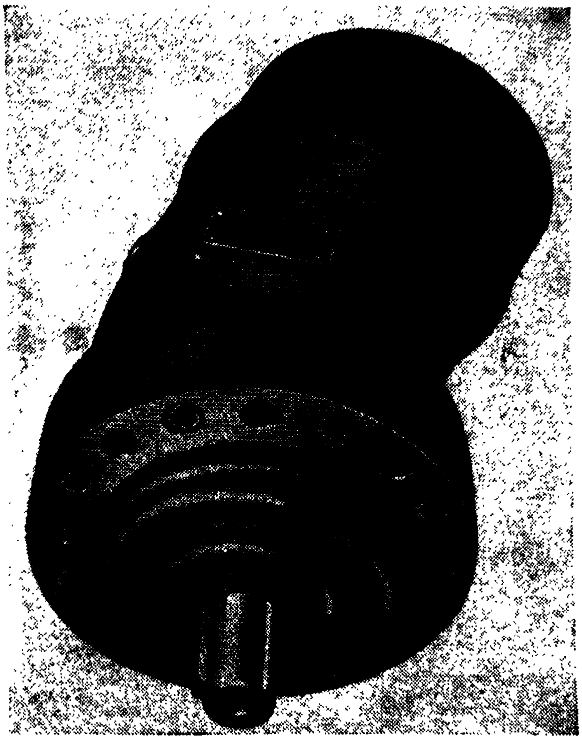


FIG. 118. Vickers axial-plunger-type motor. (*Vickers Inc., Detroit.*)

involved and not considered of sufficient general interest to be included in this text.

Commercial Horizontal-plunger Units. One of the best known of the horizontal-plunger-type motors is the Waterbury unit. These motors made with 20° fixed swash-plate angle are available in a range of sizes corresponding to the company's variable-delivery pumps. There are seven sizes ranging from 8 to 300 hp. Pressure and speed rating are the same as those of the pumps.

VICKERS INC., DETROIT. Vickers Inc. makes a very fine line of fixed-displacement horizontal-plunger-type motors. The design of the motor corresponds to that of the pump illustrated in Fig. 94 but with the port plate fixed in its angular position. This eliminates the passages and trunnions used on the pump and results in a compact assembly as shown in Fig. 118.

The motors are available in a range of sizes from 0.764 cu in. per revolution at 3,600 rpm to 15 cu in. per revolution at 1,200 rpm. There are six sizes, and in each size two tilting angles are available, $23\frac{1}{2}°$, and 30°. Operating pressures are 2,000 psi continuous and peaks of 2,500 to 3,000 for short duration. Torque efficiencies are 95 per cent, and volumetric efficiency is 92 to 97 per cent depending on conditions. Stalled torque is 90 per cent of theoretical available.

SUNDSTRAND MACHINE TOOL CO., ROCKFORD, ILL. The Sundstrand hydraulic motor is illustrated in Fig. 119. This motor operates in the following manner. Oil under pressure enters the cap section and is

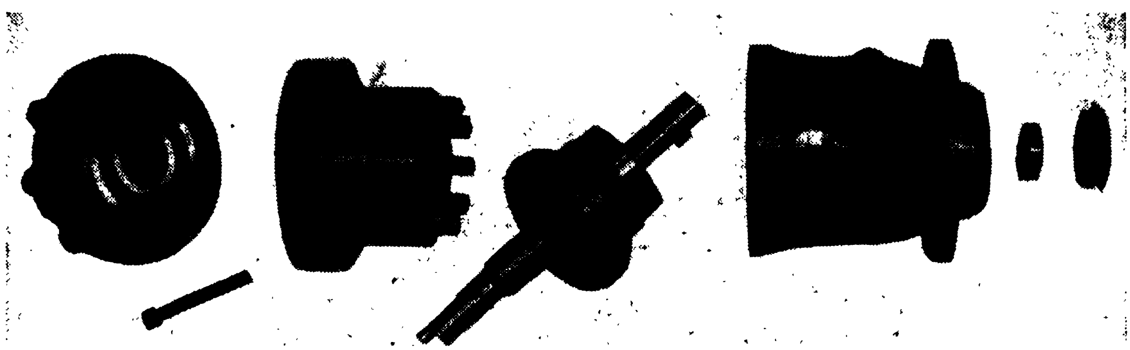


FIG. 119. Sundstrand hydraulic motor. (*Sundstrand Machine Tool Co., Rockford, Ill.*)

forced through port openings into the piston chamber. The passage of oil through the ports is controlled by a single circular valve, which is mounted on an eccentric stud on the end of the shaft. This valve does not rotate but receives a gyrating motion as the shaft turns. It is perfectly balanced hydraulically, so that there is no pressure binding. Oil pressure forces the pistons against the nonrotating wobbler. Owing to the angle at which the wobbler is inclined, the thrust of the pistons is both perpendicular and tangential to it. The resultant force is transmitted through ball and roller bearings to the wobbler plate on the shaft and imparts a rotating action to it. On the return stroke of the pistons, the piston chambers are emptied through the same port openings.

Sundstrand motors are available in single-piston design in two sizes, 2.020 and 3.340 cu in. per revolution, both at 1,200 rpm. A very unique design of double-piston motor may be supplied, giving two output speeds with different torque rating for feed and rapid-traverse operation. This motor is rated at 1.525 cu in. per revolution at 2,000 rpm and 6.530 in in. per revolution at 600 rpm. Normal operating pressure for all motors is 500 psi with peak pressures of 1,000 psi. A slight back pressure is necessary for the operation of this unit, as there is no mechanical connection between pistons and swash plate.

2. Integral Hydraulic Transmissions. Any of the pumps and motors previously described may be mounted in a common casing and combined so as to form integral transmissions or variable-speed drives. This has been done by several makers of pumps and motors, and a number of hydraulic transmissions are on the market forming compact self-contained units, suitable for driving any desired machine at a widely adjustable range of speeds and high rate of efficiency over almost the entire operating range. Operating and design principles of these combined

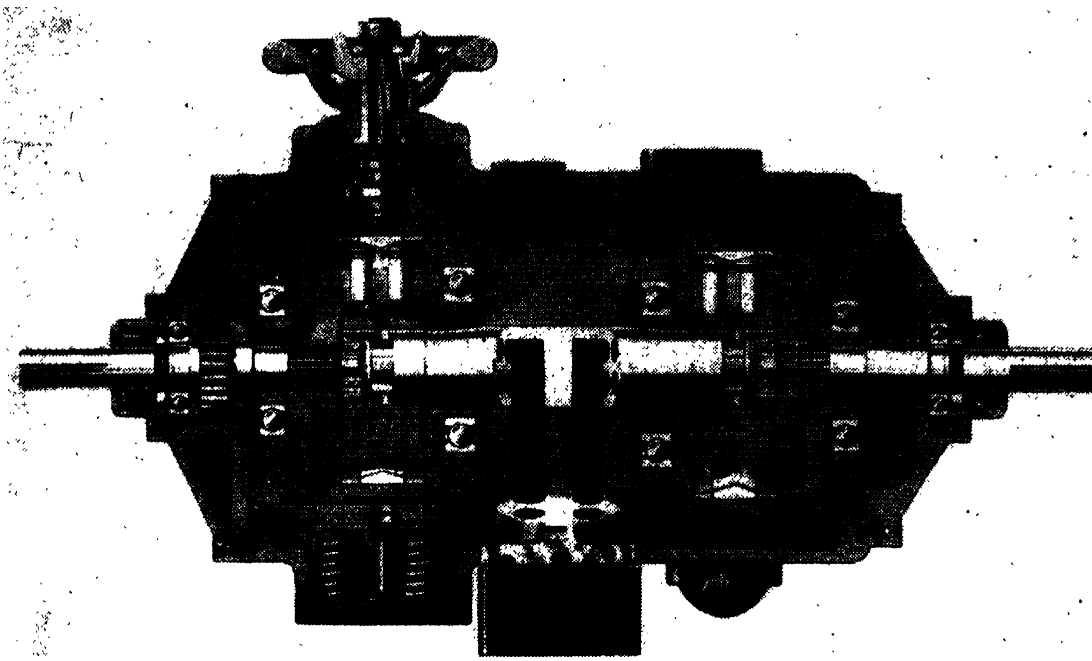
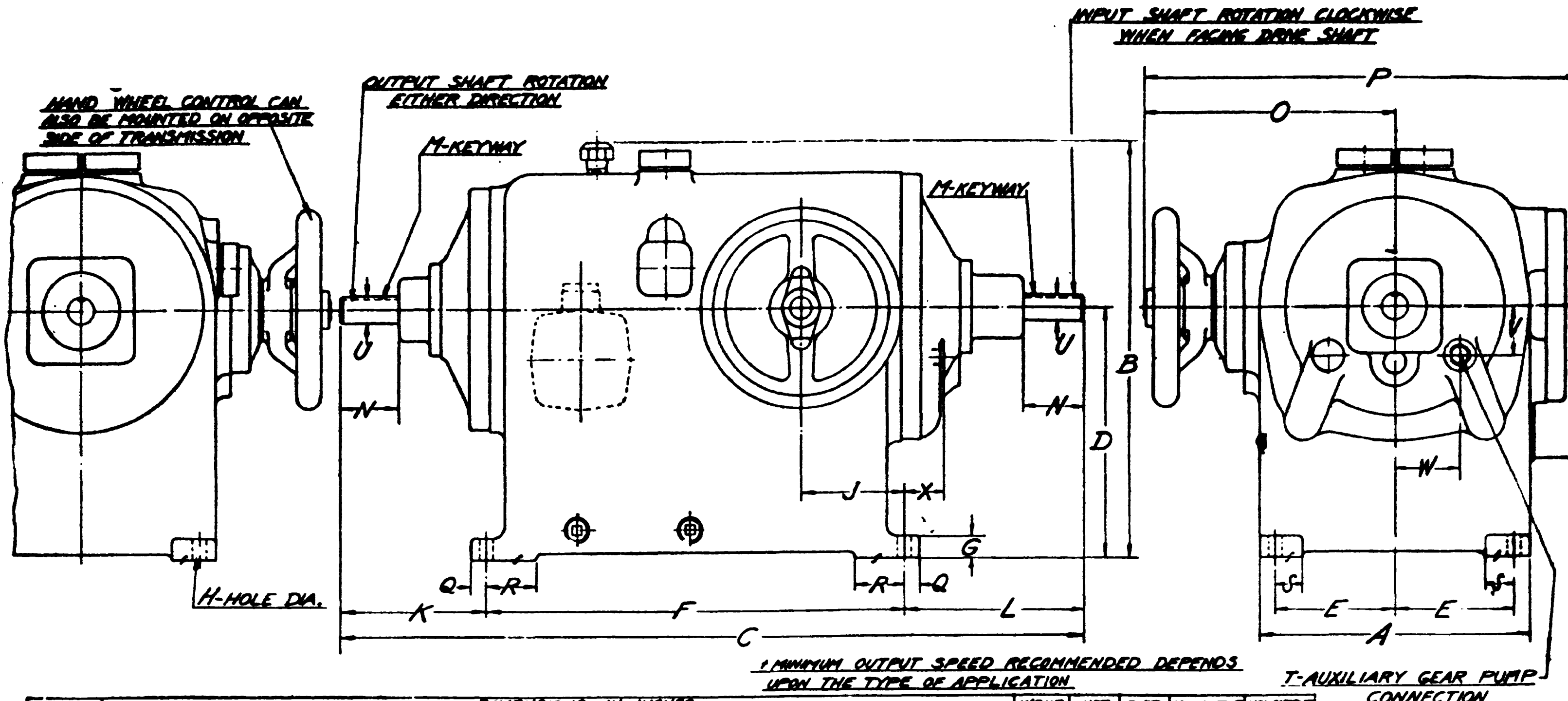


Fig. 120. Oilgear integral hydraulic transmission. (*The Oilgear Co., Milwaukee, Wis.*)

units are, of course, the same as those for individual units with connecting pipes replaced by short passages in transmission case. Means must be provided for make-up of oil from a sump or expansion tank, and suitable overload relief valves are generally supplied.

The Oilgear Hydraulic Transmission. The Oilgear integral hydraulic transmission is illustrated in Fig. 120. Both variable-delivery pump and constant-delivery motor are mounted in a common housing and have a common pintle or control valve, which is fitted in the central section of the casing. Ports are provided communicating with the pintle ports and terminating in a valve block that contains the make-up check valves. Make-up oil is supplied by a built-in gear pump, which maintains the entire hydraulic system under a head of oil at all times. A differential relief valve protects the unit from overload in either direction. Oil supply sufficient for the operation of the unit is contained in the casing.



OILGEAR TRANSMISSION SIZE	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	INPUT SPEED R.P.M.	NET WEIGHT IN LBS.	SHIP. WEIGHT IN LBS.	MIN. OUTPUT SPEED R.P.M. †	MAX. OUTPUT SPEED R.P.M.
	DIMENSIONS IN INCHES																											
DSC-22	9 1/8	14 15/16	29 3/8	9	3 13/16	16 3/8	1	9/16	3 3/4	6 1/2	6 1/2	1/4 x 1/8	2 1/2	9 3/8	16 3/16	5/8	2	1 1/4	1/2	1.0625 1.0620	2 1/8	2 1/4	1 5/8	1140	320	380	5 to 30	1090
DSC-44	10 3/4	16 7/8	30 5/8	10	4 3/4	16 13/16	1 1/8	9/16	4 1/16	6 15/16	6 1/2	1/4 x 1/8	2 1/2	10 1/8	17 5/16	5/8	2	1 1/8	1/2	1.0625 1.0620	2 13/16	2 19/32	1 5/8	1140	420	500	5 to 30	1090
DSC-88	11 3/4	18 5/16	34 3/4	11	5 1/4	20 3/8	1 1/8	9/16	4 5/8	7 5/16	7 5/16	1/4 x 1/8	3	11 3/16	19 3/8	5/8	2	1 1/4	1/2	1.2500 1.2495	2 9/16	2 3/4	1 7/8	1140	550	640	5 to 30	1090
DSC-1212	13 1/8	20 1/16	37 1/2	12	5 3/4	23 1/2	1 1/4	11/16	5 3/8	7	7	3/8 x 3/16	3 1/4	11 3/4	21 7/16	13/16	2 1/4	1 7/16	1/2	1.4375 1.4370	2 3/4	2 27/32	1 1/2	1140	835	910	5 to 30	1090
DSC-2020	14 1/8	21 7/16	43 3/4	13	6 1/4	28	1 1/4	11/16	6 3/4	7 3/8	7 7/8	3/8 x 3/16	3 3/4	12 5/8	23 1/2	13/16	3	1 3/8	1/2	1.624 1.625	3 1/4	3 1/4	1 11/16	1140	1025	1225	5 to 30	1090

FIG. 121. Dimensions of Type DSC transmissions. (*The Oilgear Co., Milwaukee, Wis.*)

Five sizes are available in this integral transmission, ranging from 2.8 to 26.1 input hp. Minimum output speeds are from 5 to 30 rpm. Like all fixed-stroke motor transmissions, the unit is a constant-torque transmission, horsepower varying as speed. Figure 121 shows the dimensions of the transmission.

Transmissions may also be supplied with flange-mounted motors and built-in speed reducers for a 3:1 reduction of the motor speed. Any of the controls described in Chap. VII may be furnished with the unit. Figure 122 shows a unit with remote stroke adjustment by electric gear reduction

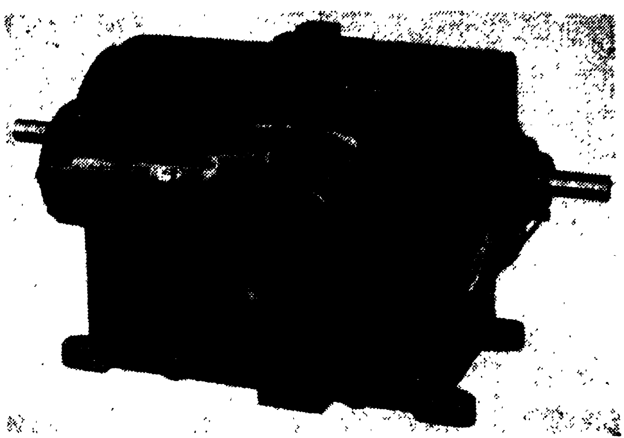


FIG. 122. Transmission with remote control by electric pilot motor. (*The Oilgear Co., Milwaukee, Wis.*)

motor. Efficiency of the unit is excellent. Figure 123 shows efficiency curves of a DSC 2,020 unit as function of output speed at full load rating. The units operate with about 1,200 psi normal pressure with permissible peak loads of 150 per cent.

The company has recently placed on the market a newly designed integral hydraulic transmission, which is illustrated in Fig. 124. The unit is of compact design incorporating a number of new features. Slanted rolling pistons are employed bearing against hardened reaction rings. A flat plate valve, replacing the conventional pintle, is used. Port-plate separating force is balanced by auxiliary pistons mounted in the port plate instead of by the plunger reaction customarily utilized in axial plunger units.

The unit weighs 90 lb and develops up to 3 hp at 1,600 rpm output speed. Standard controls supplied by the company for its large pumps may be furnished to control output and pressure of the pump in any desired manner. The transmission may also be supplied in separate pump and motor units and connected by pipe lines.

The Waterbury Hydraulic Transmission. The Waterbury transmission is, to the author's knowledge, the oldest commercially practical hydraulic

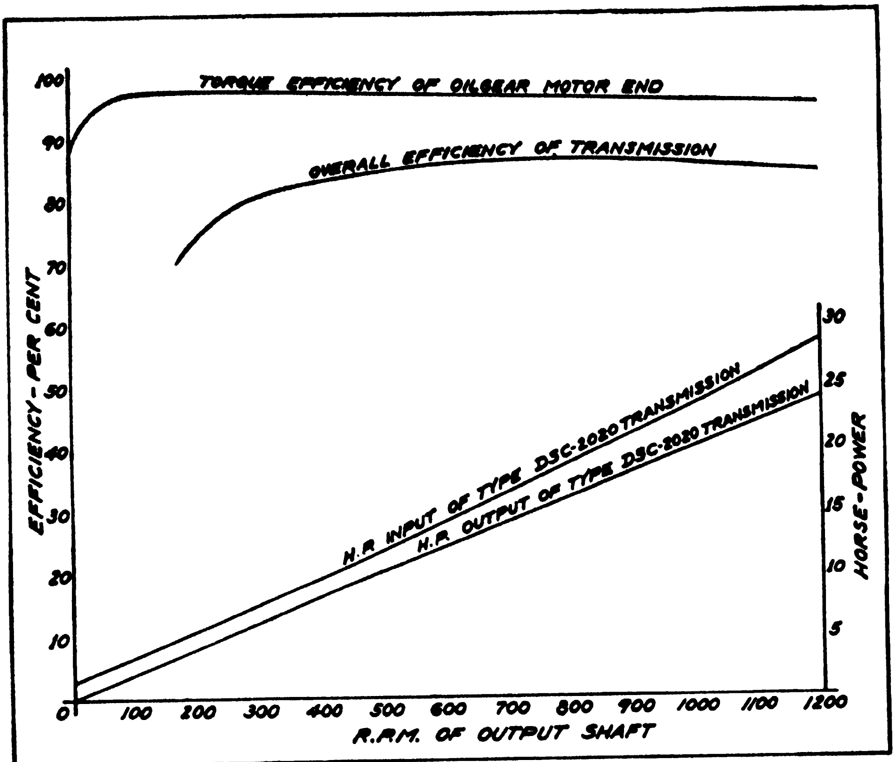


FIG. 123. Performance curves at full load rating of Oilgear DSC 2,020 transmission. Operating temperature: 120°F. Oil viscosity 250 SSU at 100°F. (*The Oilgear Co., Milwaukee, Wis.*)

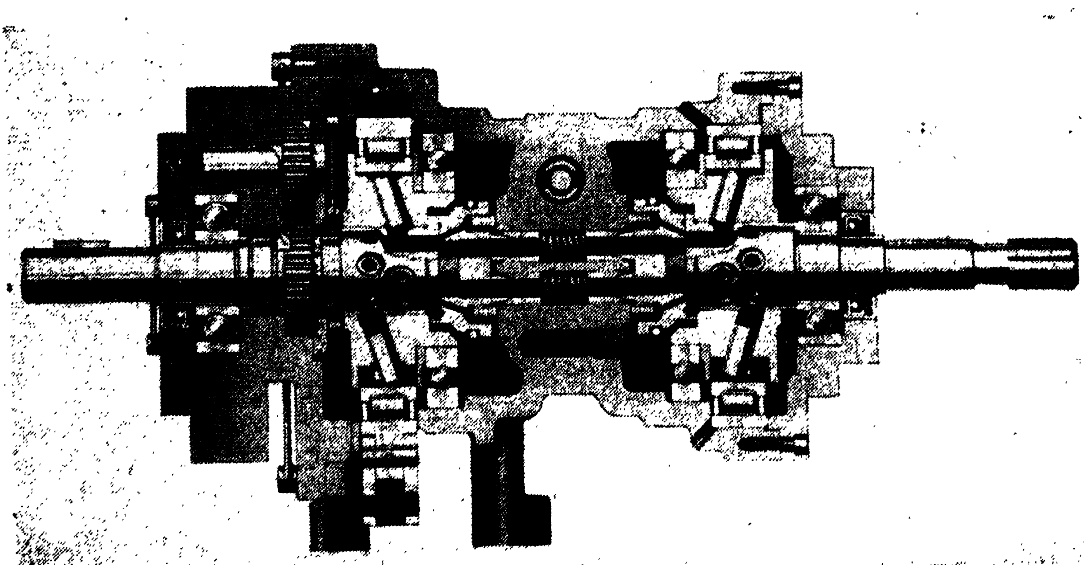


FIG. 124. **New design of Oilgear integral transmission.** (***The Oilgear Co., Milwaukee, Wis.***)

transmission. For this reason the following historical sketch, which has been supplied through the courtesy of the Waterbury Tool Co., is undoubtedly of interest to the reader.

HISTORICAL BACKGROUND OF THE WATERBURY TRANSMISSION. Initial efforts to invent and develop a hydraulic transmission date back to a time prior to 1900. The two earliest attempts were recorded in 1892 and 1900, but were not practically capable of development and the work on them was dropped. In 1901 or 1902, Harvey Williams, formerly an instructor at Cornell University, began work on what was to become the Waterbury transmission, and interested H. G. Hoadley, owner of the Waterbury Tool Company, in the manufacture and development of his transmission. The association of the two men in this venture was not something new, but was a continuance of relations that started when Hoadley was a student at Cornell. Contact between the two men had continued after Hoadley's graduation, and, in fact, the Waterbury Tool Company was started in 1898 to manufacture a ratchet drill conceived by Williams. William's initial work on the transmission was accomplished with a view toward use in the automobile, but the model manufactured and developed by the Waterbury Tool Company had certain defects that seemed to preclude its use commercially, and the matter was dropped for the time being.

In the meantime, Williams, who was affiliated with the Bureau of Ordnance as an engineer, interested the Navy in the transmission when he found that they were attempting to locate a substitute for the Ward Leonard system of gun control. Investigations were begun anew with a view to overcoming the difficulties inherent in earlier designs, and a complete set of new drawings was completed.

Reynold Janney, an engineer working for the Waterbury Tool Company, did most of the development work, and under his direct supervision, two machines were rushed through manufacture with all possible speed. After thorough testing, both at the Waterbury Tool and in university laboratories, the machines were taken in 1905 to Washington for demonstration and further testing to determine whether or not they were suitable for use in elevating guns on battleships.

The result of this was an order for a machine to elevate the 12-in. guns of the "Virginia." This machine was installed in 1906, and at the same time, another unit was ordered for turret training in the battleship "Illinois." From that time on, larger orders were issued by the government, and all guns and turrets were equipped as rapidly as manufacture and the limitations of ship design would permit.

Since that time, the naval applications have spread until the units are now employed not only in elevating and training applications but on

windlasses, boat and airplane cranes, steering gears, fuel-oil pumps, ammunition and powder hoists, and rammers.

The units have also had numerous commercial applications since there has been considerable demand for a unit that would provide smooth, sensitive, and accurate control. A partial list of such applications is as follows:

Paper-mill drives
Machine-tool applications
Glass-tubing machines
Wire-rope machines
Textile machinery
Billet gougers
Flying shears for steel mills
Printing presses
Wiredrawing machines

The Waterbury Integral Transmission. The Waterbury integral transmission is shown in Fig. 125. The unit operates with the case filled with

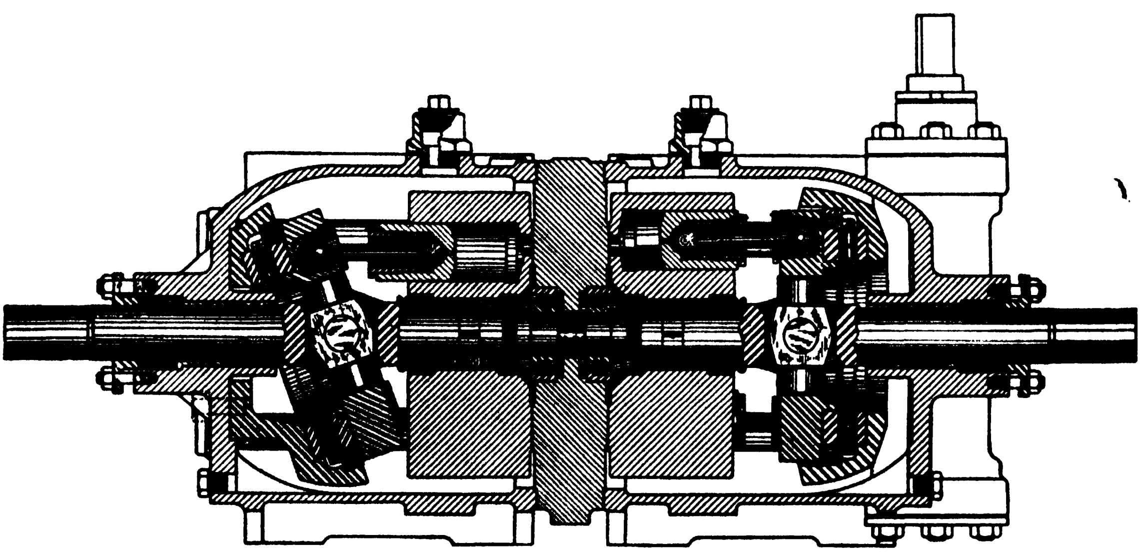


FIG. 125. Waterbury transmission. (*The Waterbury Tool Co., Waterbury, Conn.*)

oil and is provided with an expansion tank, shown in Fig. 125. Relief and check-type replenishing valves are mounted in the valve plate. The individual units, pumps and motors, are identical with those described before. The same range of sizes is available as in the separate pumps and motors. The outstanding characteristic of this unit is its extreme speed range. Output speeds as low as 1 rpm are possible. This is due to the excellent workmanship, close fit in the pistons, and the hydraulically balanced valve plate, resulting in minimum leakage. Efficiency curve of a unit is shown in Fig. 126.

The Vickers Variable-speed Transmission. Vickers variable- and constant-delivery pumps may be supplied mounted in a common housing to form integral transmissions. They may be furnished in any of the following combinations:

1. Variable-delivery pumps and constant-delivery motors, resulting in constant-torque variable-horsepower units. At a given loading or hydraulic pressure, horsepower varies as speed, and torque is constant over the entire speed range.

2. Constant-delivery pumps and variable-delivery motors resulting in a constant-horsepower transmission. At a given hydraulic pressure,

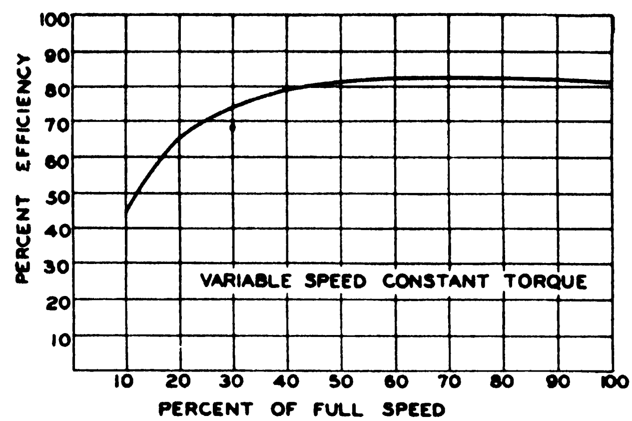


FIG. 126. Efficiency curve of Waterbury transmission. (*The Waterbury Tool Co., Waterbury, Conn.*)

horsepower is constant between a minimum speed, determined by the geometric capacity of the motor at full-stroke setting, and a maximum speed at reduced-motor-stroke setting dictated by design limits (bearing speeds, etc.). The torque varies inversely as the speed. These units are generally limited to a speed range of 3:1.

Variable-delivery pumps and variable-delivery motors may be combined to obtain any desired combination of torque and horsepower. Very wide speed range may be obtained with this arrangement. Adjusting the motor for its full displacement and decreasing pump stroke will decrease horsepower with decreasing speed and constant torque until the practical minimum, determined by leakage losses, etc., is reached. On the other hand, decreasing motor stroke at full pump displacement will increase speed to the practical maximum dictated by design considerations, while torque varies inversely with speed and horsepower remains constant.

Vickers transmissions are available in a wide variety of sizes and combinations from 2 to 500 hp. Minimum speed recommended is 50

rpm with maximum output speeds as high as 3,600 rpm for small units. The design of the unit is illustrated in Fig. 127.

The Sundstrand Oil-power Transmission. The transmission is of the variable-horsepower variable-torque type and consists of two variable-

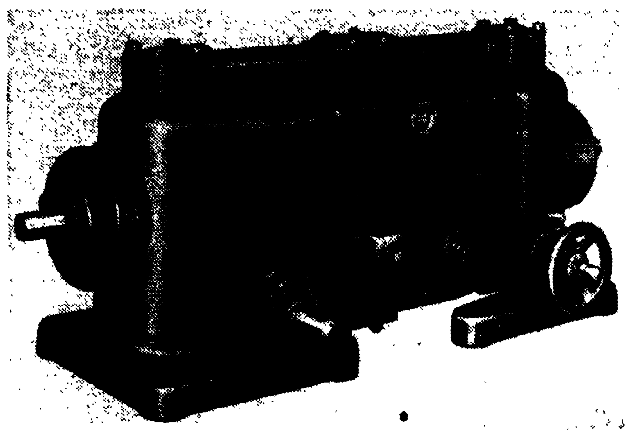


FIG. 127. Vickers hydraulic transmission. (*Vickers Inc., Detroit, Mich.*)

displacement units of a design similar to the company's hydraulic motor described previously. Both units are mounted in a housing that must be installed on or provided with a separate oil reservoir. A small pilot

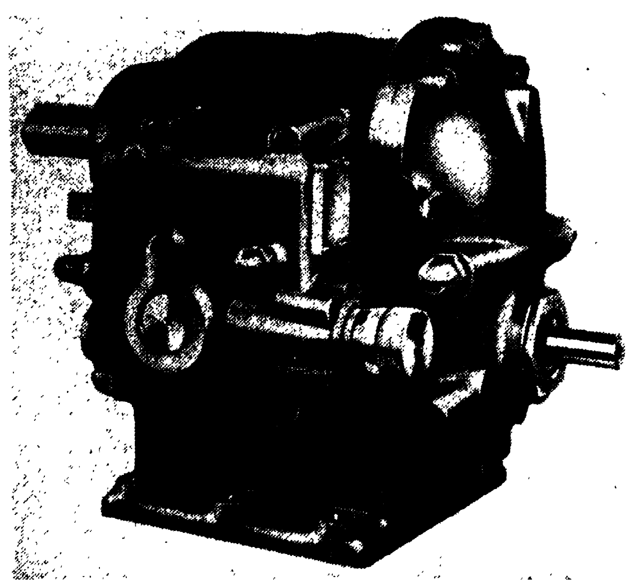


FIG. 128. Sundstrand hydraulic transmission. (*Sundstrand Machine Tool Co., Rockford, Ill.*)

pump supercharges the transmission, provides spray lubrication, and furnishes pilot pressures for the hydraulically actuated stroke adjustment. Stroke adjustment is made by suitable adjustable stops, and the transmission may be started, stopped, and reversed by a control valve that directs the flow of pilot pressure fluid to the stroke-adjusting plungers. The transmission is illustrated in Fig. 128.

The unit develops $1\frac{1}{2}$ hp at 500 to 2,400 rpm. Minimum speed is 10 rpm. Operating oil, 3 gal recommended, should have viscosity of 315 SSU at 100°F.

3. Reciprocating Motors (Hydraulic Cylinders). The oldest and still most frequently used application of hydraulic power is the hydraulic ram. In their modern form, hydraulic rams power machinery from heavy hydraulic presses to small grinders, exerting pressures from thousands of tons to a few hundred pounds. Hydraulic rams may be of the single- or double-acting type. The design of the former, illustrated in Fig. 129, permits application of hydraulic pressure in one direction only and requires separate retraction cylinders to return it to its starting position. These cylinders are made in all sizes from a fraction of an inch to 60 in. and larger.

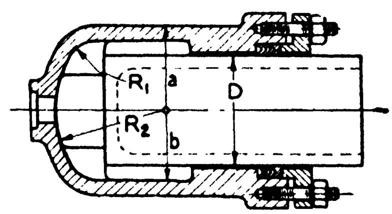


FIG. 129. Single-acting hydraulic cylinder.

Materials and Design. Hydraulic cylinders may be made from alloy or high-tensile cast iron for lower pressures (up to 1,000 psi) and cast steel, forgings, or steel tubing for higher pressures. Cast-steel cylinders are made with dome-shaped heads, as shown in the illustration. For forged or welded cylinders this is not always feasible. Radii R_1 and R_2 are two third and one third of D, respectively. Wall thicknesses may be computed as thick-walled cylinders with internal pressure. The stress in a hydraulic cylinder is triaxial, the principal stresses at the inside diameter (point of maximum stress) being

$$s_1 = \frac{a^2 + b^2}{a^2 - b^2} p \tag{31}$$

$$s_2 = -p \tag{32}$$

$$s_3 = \frac{b^2}{a^2 - b^2} p \tag{33}$$

For general hydraulic applications, s_2 and s_3 are not of sufficient magnitude to affect the resultant stress materially, and s_1 may be used as criterion. Allowable stresses depend upon type of material and application, ranging from 6,000 psi for alloy cast-iron and 12,000 for steel to 85,000 for highly stressed alloy-steel cylinders for aircraft. Dome-shaped heads are customarily made the same thickness as the walls. Flat cylinder heads must be figured as flat plates, semirigidly held. Caution is indicated to allow for at least a small radius in the corner for flat heads integral with cylinders.

Rams of less than 4-in. diameter should be steel and should be carburized, hardened and ground, or chrome plated. Rams of 4-in. diameter

and larger may be made from semisteel, turned and polished. Packings today are almost exclusively the self-tightening or automatic type, either the V or O ring. V rings are preferable for larger sized rams and higher pressures. For steel rams, hardened and ground, a homogeneous type of synthetic packing is recommended. This packing is available in sizes from 1/8 to 15 in. ID. One ring of packing is recommended for each 500 psi pressure to be held, but never less than two rings should be used. The widths given in Table I are recommended.

TABLE I

Ram diam.	Width	Stack height
Up to 1/2	3/16	0.083
1 1/4	1/4	0.083
2 1/2	5/16	0.140
3 7/8	3/8	0.156
5 1/2	7/16	0.197
15	1/2	0.197

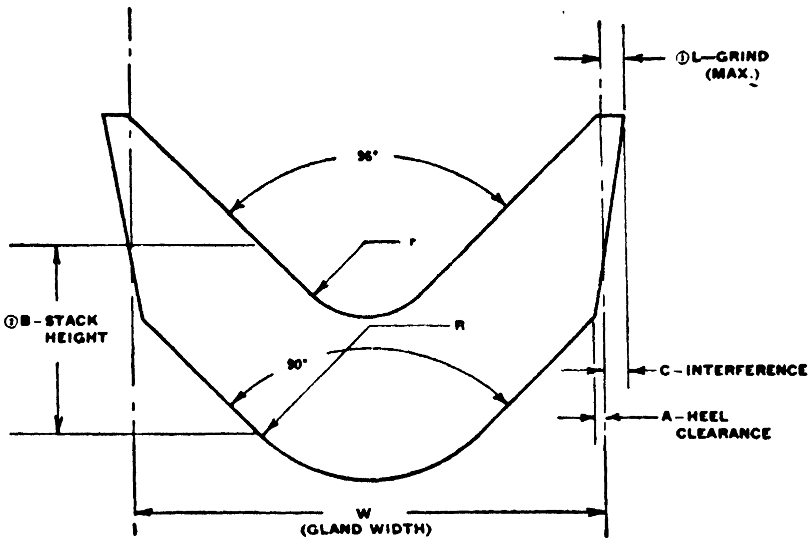
Figure 130 shows an enlarged cross section of the Linear V packing. Complete sets of V rings and fabric-reinforced adapters are furnished by the packing manufacturer. Where metal adapters are desired, they may be made from aluminum or bronze and must be designed to fit the packing. To this end, drawings may be secured from the packing manufacturer showing top and bottom rings to fit their particular make of packing.

The clearances and tolerances given in Table II are recommended.

TABLE II

Outside diameters	Tolerance, nominal	Inside diameters	Tolerance, nominal	Resulting clearance	
				Min.	Max.
Rams or piston rods, up to 2 1/2 in.	−0.002 −0.004	Gland bore, up to 2 1/2 in.	+0.002 +0.004	0.004	0.008
Over 2 1/2 in.	−0.003 −0.005	Over 2 1/2 in.	+0.003 +0.005	0.006	0.010
Gland OD, up to 3 1/4 in.	−0.002 −0.004	Packing chamber ID, up to 3 1/4 in.	+0.002 +0.004	0.004	0.008
Over 3 1/4 in.	−0.003 −0.005	Over 3 1/4 in.	+0.003 +0.005	0.006	0.010

Female adapter rings should be fitted with the same clearance as glands. Male adapters may have $\frac{1}{64}$-in. clearance on inside and outside diameters.



NOTES

① "L" (MIN.) IS .006 FOR ALL DASH NUMBERS

② "B" (STACK HEIGHT) IS MEASURED AT POINT OF TANGENCY OF RADIUS "R" & 90° ANGLE.

DASH NO.	NOM. SIZE	"W" (REF.)	"A" (REF.)	"B" ±.010	"C" (REF.)	"R"	"r"	"L" (MAX.)
1-7	3/16	.1905	.006	.083	.014	1/16	1/32	.020
8-24	1/4	.2530	.006	.083	.012	1/16	1/32	.020
25-35	5/16	.3155	.007	.140	.024	7/64	3/64	.035
36-47	3/8	.3780	.009	156	.030	1/8	3/64	.035
48-55	7/16	.4415	.010	.197	.037	5/32	1/16	.035
56-60	1/2	.5040	.010	.187	.032	5/32	1/16	.035

FIG. 130. Cross-sectional view of V-ring packing. (*Linear Inc., Philadelphia, Pa.*)

The following notes are quoted from the design handbook of Linear Inc. with their permission:

1. *Adapters.* All V-ring adapters should be made to Linear drawings. Much of the sealing and endurance qualities of V-ring packings depend on the support given by the adapters, especially the female, and precision methods of manufacture consistent with reasonable economy should be employed.

2. *Bearing Areas.* In the determination of rod or piston-head bearing lengths, V-ring packings and male adapters should not be considered as a bearing area. Female adapters, however, may be so considered, provided that recommended clearances are held.

3. *Orientation.* On piston installations where the pressure acts from both directions, such as double-acting cylinders, opposing sets of V-ring packings should always be installed so that the sealing lips face toward the pressure and away from each other. The female adapter in all cases should be located adjacent to a fixed or rigid part of the piston so that no pressure loads will be transmitted to the other set of oppositely facing rings.

4. *Installation Obstructions.* All packing installations should be designed so that it is not necessary to install the rings over or through threaded surfaces or other sharp projections, without permitting adequate clearance (approximately 10 per cent of free cross section of packing) between the diameter of the packing rings and the projection. Such precautions will eliminate the danger of damaged seals causing failure of the unit in operation.

5. *Multiple Assembly.* From the operational standpoint, installation of each packing ring individually is the most satisfactory method of ensuring the best service. However, several methods have been devised by various manufacturers by which a complete set of packing may be installed as a unit. It should be warned, however, that any such methods should be carefully and extensively tested before they are adapted as standard practice, as there have been numerous cases of packing failure directly attributable to this cause.

6. *Mechanical Loads.* Bottoming loads should never be transmitted to the packings. V-ring packings are designed to seal fluid pressure only and not to act as shock absorbers or bumpers.

7. *Finishes.* All cylinder and valve bores should have a smooth, ground and lapped or honed finish on all surfaces through which the packing moves. All external surfaces should have a polished hard chrome surface or equivalent. Recommended finishes are as follows:

Piston rod or cylinder bore: 5 to 15 micro-inches rms

Other surfaces against which packing must seal statically: 20 micro-inches rms (maximum)

Needless to say, rougher surfaces have been and may be used, but it can be generally stated that the expected life of the packing depends directly on the surface finishes of the moving parts.

Packing glands may be made from cast steel, forgings, or steel plate. When used with steel rams, glands should be bronze bushed. Studs should be conservatively stressed. Self-locking nuts are recommended for packing-gland applications. For cast-iron rams the homogeneous type of packing is not suitable. For this purpose, V packing consisting of Neoprene-impregnated cotton duck is recommended. This packing is supplied in sets complete with top and bottom adapter. Again one ring

is required for each 500 psi. Greater liberty can be taken with clearances and tolerances in using this type of packing. Clearances of rams and glands should be from 0.01 to 0.03 according to size. Widths range from ½ to ¾ and wider according to ram size.

Double-acting Rams. This type of cylinder has gained considerable popularity since the introduction of oil hydraulics. Double-acting cylinders have been made in sizes as large as 48 in. and for any pressure encountered in oil hydraulic practice. Numerous constructions have made their appearance, and a number of firms have available standardized sizes and designs which may be purchased at great economy and installed in hydraulically actuated machines. Designs vary greatly according to requirements and preference of the firms manufacturing them.

Figure 131 shows the design of a large cylinder for oil-pressure presses and similar hydraulic machinery. This design, used for sizes of about 8 in. and up, features cast-steel cylinder and semisteel ram with cast-iron piston rings and composition V packing. The bore of the cylinder should be machined smoothly and polished. Excellent results are obtained with honing cylinder bores where feasible. Standard cast-iron piston rings are used for sealing the cylinder heads. Number of rings varies from a minimum two for very small pistons up to eight for the very largest. Fit allowances and tolerances for pistons and bores should follow the recommendations in Table I, Chap. V. A very good, reasonably oiltight fit may be made with this arrangement.

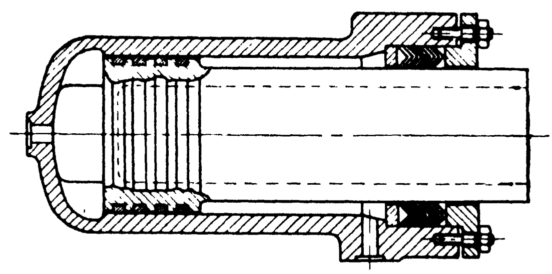


FIG. 131. Double-acting cylinder.

Differential areas depend, of course, upon the load required for the retraction stroke, traveling speed, etc. It is well to remember that there is a lower limit below which one cannot go in determining this area. When the force obtainable at any given pressure on the differential area is less than the friction created on the piston rings at the same pressure, then the ram will not move at any pressure. Figures determined by practical experience are given in Table III.

TABLE III

Piston Diam. D, In.	*Minimum Difference between Piston and Rod Diam., D − d, In.*
Up to 6	¼
Up to 20	½
Up to 32	¾
Up to 40	1

A tapered counterbore at the mouth will facilitate inserting the rings into the cylinder bore. With a small shoulder left for the bottom packing-support ring, the width of the packing is established. Packing wider than $1\frac{1}{4}$ in. is not recommended, and in case of large differentials, a separate cylinder head must be provided. Smaller differential cylinders may be made with cast-iron cylinders and steel piston rods. Hardening and grinding or chrome-plating rods is recommended. In this case, homogeneous packing may be used. Cylinders of this type are generally made with separate heads, and the joint between head and cylinder must be sealed with a suitable gasket. For hydraulic work, either fully retained compression gaskets or O rings may be used. Compression gaskets should be made from soft, fully annealed copper and be from $\frac{1}{8}$ to $\frac{1}{4}$ in. wide and $\frac{1}{16}$ to $\frac{1}{8}$ in. thick. Goetze multiseal gaskets are excellent for this purpose and should be used in somewhat wider widths.

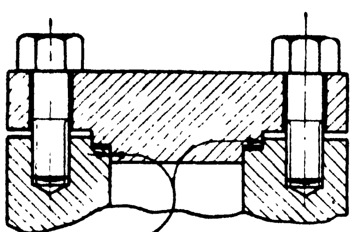


FIG. 132. Design of cylinder head with compression gasket.

Figure 132 shows the design of a cylinder head with this type of gasket. Care should be taken to make the cylinder head heavy enough in section to prevent its being sprung when the screws are tightened. Socket-head cap screws are becoming increasingly popular for this application. Bolts and cap screws should be spaced closely and used with moderate stresses to allow for tightening with heavy preloads. This type of seal relies for tightness entirely on the gasket pressure developed by the cap screws, and it is often difficult to produce a leakproof seal. These objections may be overcome by the use of O-ring gaskets, which are becoming increasingly popular. O rings, as the name implies, are rings of circular cross section that are inserted in a counterbore with slight "squeeze" or compression. The hydraulic pressure forces these rings outward and effectively seals the crack between head and cylinder. The cylinder head is bolted to the flat surface of the cylinder, producing a controlled initial compression of the O ring. Great care must be taken in the machining of the counterbore; there must not be any tool marks or rough places and particularly no pinholes or small cracks, which might cause the O ring to blow out. An O ring will blow through a crack as narrow as 0.008 in. O rings may be supplied in almost any diameter and a number of standard widths.

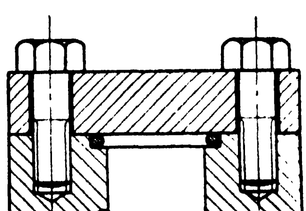


FIG. 133. Design of cylinder head with O-ring gasket.

Linear standards call for the O-ring-gasket dimensions given in Table IV.

TABLE IV

OD	Width	Depth of counterbore	Min. squeeze
Up to ½	0.070 ± 0.003	0.057	0.010
15/16	0.103 ± 0.003	0.090	0.010
Over 15/16	0.139 ± 0.004	0.123	0.012

Outside diameter of counterbore should be made equal to or somewhat less than the outside diameter of O ring, if it is desired to prevent the ring from falling out in assembly.

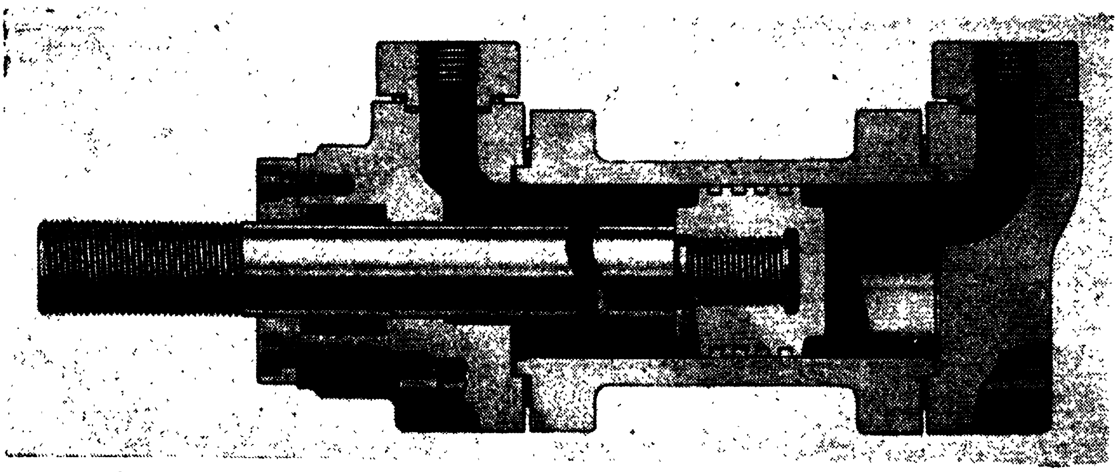


FIG. 134. Double-acting cylinder. (*The Oilgear Co., Milwaukee, Wis.*)

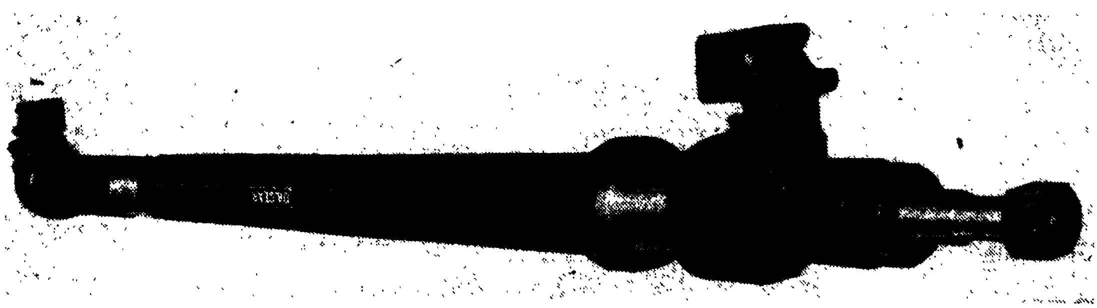


FIG. 135. Hydraulic-planer cylinder. (*The Oilgear Co., Milwaukee, Wis.*)

Figure 134 shows the design of double-acting cylinder developed by the Oilgear Co. Figure 135 shows a very long cylinder used for a planer drive made by the same company.

The design of the Gerotor May Corp. is shown in Fig. 136. The cylinder is made from steel tubing, and the method of attaching the flanges is interesting: O rings are used as seals rather than gaskets. This

application of O rings will be discussed farther on in this section. The piston rod is threaded into the piston and locked with a lock nut. Another method of securing the piston rod, which is preferred by the author, is a force fit in the piston against a shoulder in the rod with thread and a castle nut to hold the piston on the rod. Oil leakage past the press fit is infinitesimal.

Double-acting Cylinders with Packed Pistons. For the majority of applications, regular cast-iron piston rings are very satisfactory for pack-

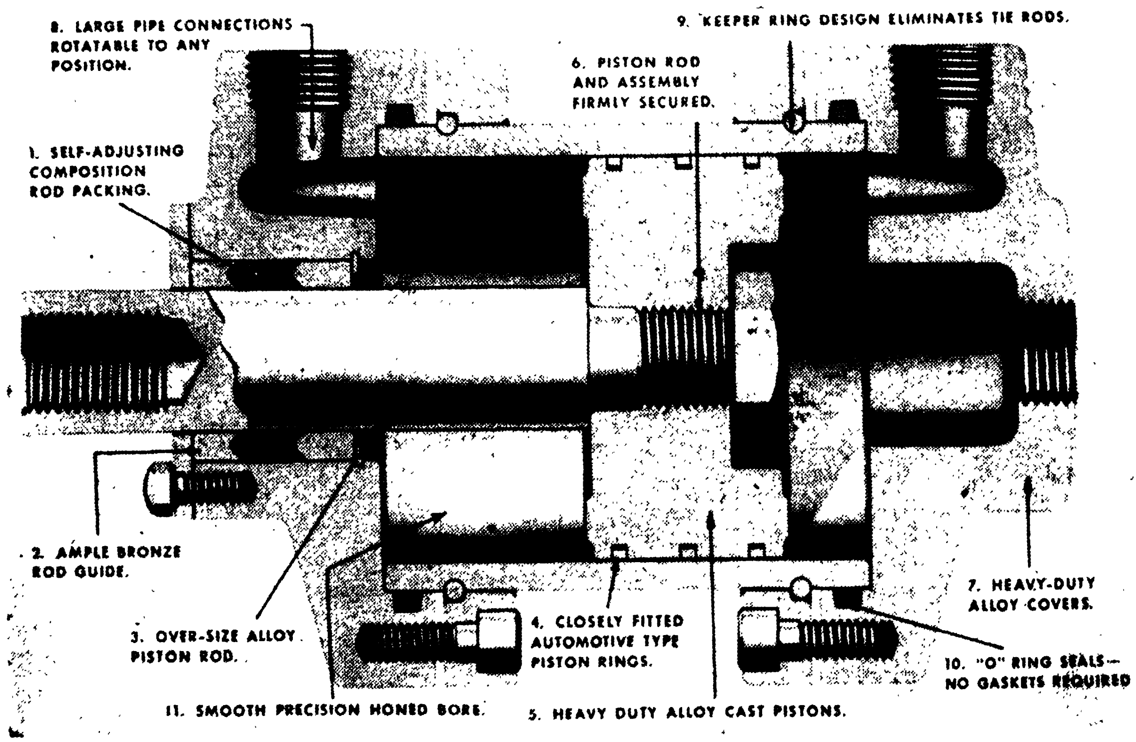


FIG. 136. Double-acting cylinder. (*Gerotor May Corp., Baltimore, Md.*)

ing the pistons. If bores and pistons are properly machined to correct tolerances, very little leakage will be experienced. There are cases, however, such as in aircraft cylinders, where absolutely no leakage can be tolerated. In such cases, pistons may be packed with V-type packings. The design of such a cylinder is shown in Fig. 137. This design shows an aircraft type of hydraulic cylinder, but it may be equally well adapted for machine-tool and other hydraulic applications. Rules given in this chapter for finishes, clearances, etc., on V rings for ram and rod applications apply to piston applications as well. Flash chrome plating of cylinder bores is recommended where packings are used on pistons. None but steel cylinders are suitable for this application.

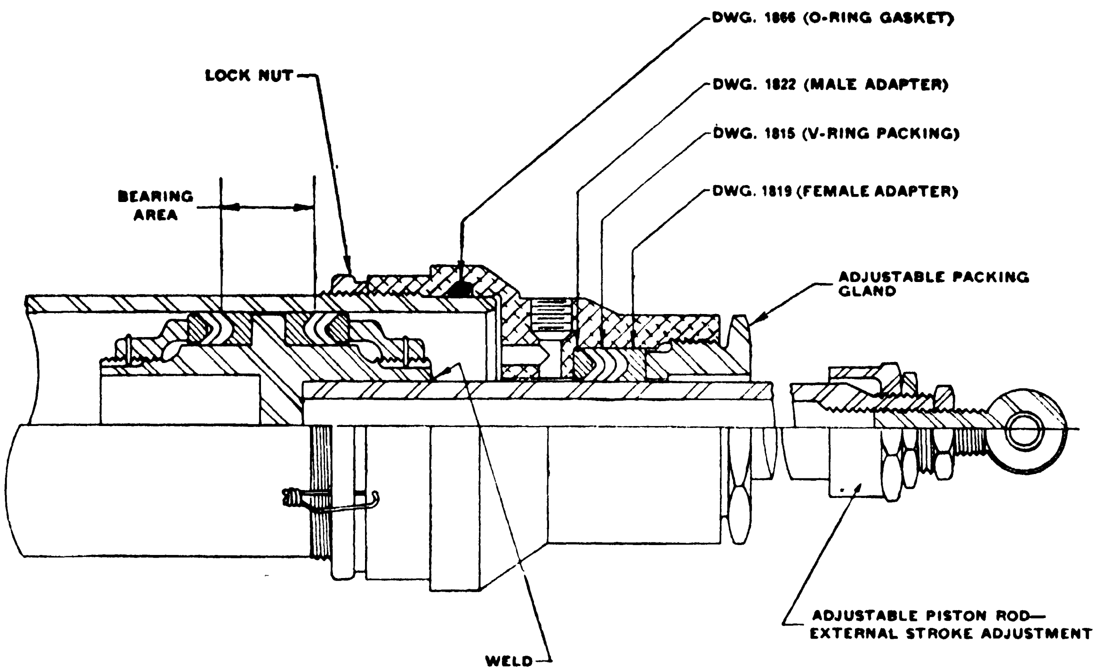


FIG. 137. Typical hydraulic actuating cylinder with V-ring packings. (*Linear Inc., Philadelphia, Pa.*)

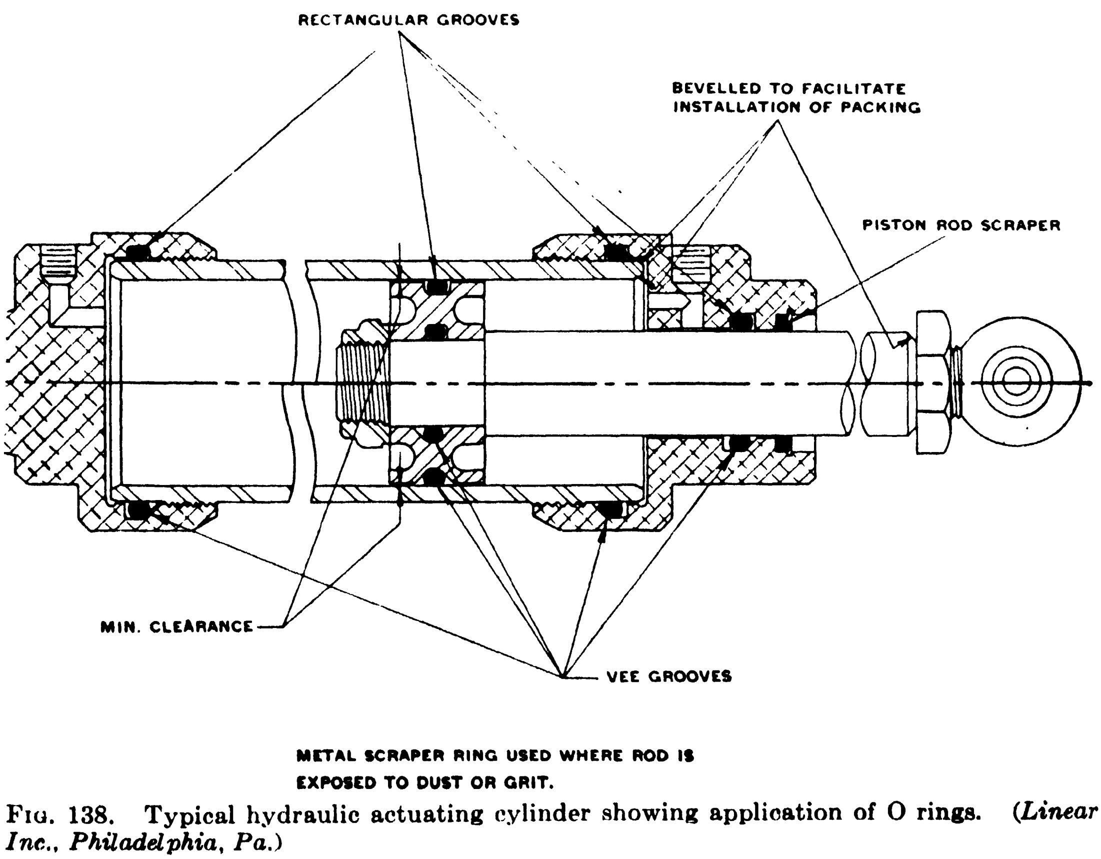


FIG. 138. Typical hydraulic actuating cylinder showing application of O rings. (*Linear Inc., Philadelphia, Pa.*)

The O-ring Packing. One of the most outstanding developments in the art of hydraulics is the invention of the O ring by Niels A. Christensen (U.S. Patent 2,180,795). This type of packing has revolutionized aircraft hydraulics and has subsequently invaded the commercial field, where it has established itself by its low cost, simplicity, and ease of installation. O-ring applications may be divided into static and dynamic applications. An example of static application is the O-ring gasket, which was described earlier in this section. Another static application is shown in Figs. 136 and 137. Dynamic applications cover rod and piston packings. A typical hydraulic cylinder showing O rings applied on piston, rod, and cylinder-head seals is shown in Fig. 138.

TABLE V. SIZES OF STANDARD O-RING PACKINGS, INCHES*

Width	ID	OD	Actual width	Min. squeeze	Groove width		
					Rectangular	V 45°	Bottom radius
1/16	Up to 3/8	1/8 + ID	0.07 ± 0.003	0.010	0.093	9/64	1/32
3/32	Up to 3/4	3/16 + ID	0.103 ± 0.003	0.012	0.140	7/32	3/64
1/8	Up to 1 1/2	1/4 + ID	0.139 ± 0.004	0.012	0.187	19/64	1/16
3/16	Up to 4 1/2	3/8 + ID	0.210 ± 0.005	0.017	0.281	15/32	3/32
1/4	Up to 16	1/2 + ID	0.275 ± 0.006	0.029	0.375	39/64	1/8

* Data supplied by Linear Inc., Philadelphia, Pa.

TABLE VI. RECOMMENDED CLEARANCES AND TOLERANCES, INCHES

Cylinder bores, nominal		Piston and gland groove diam.	Rod diam., nominal	
Up to 15/16	±0.001	±0.001	Up to 3/4	−0.001 −0.003
Up to 3 1/4	+0.002 −0.000	±0.001	Up to 2 7/8	−0.002 −0.004
Up to 4 7/8	+0.003 +0.001	±0.001	Up to 16 1/2	−0.003 −0.005
Up to 16	+0.004 +0.002	±0.001		

Tables V and VI give sizes and recommended clearances. Grooves may either be rectangular or V type. Rectangular grooves should be made with approximately 1/32-in. bottom radii and all sharp corners broken. Care must be taken to bevel entrance for packings and avoid

all sharp corners, tool marks, nicks, or scratches that might mar or cut packings. Clearance between pistons and cylinders should be minimum to prevent extrusion of O rings.

Determination of Piston and Gland Groove Diameters[1]

1. Piston groove diameter. To obtain piston groove diameter:
   a. Take bore diam.
   b. Add positive tolerance
   c. Subtract twice min. width
   d. Add twice min. squeeze
   e. Result: min. piston groove diam.

Example:
a 1.875
+*b* 0.002
1.877
−*c* 0.410
1.467
+*d* 0.034
1.501 min. piston groove diam.

2. Determination of range of squeeze

a. Take max. bore diam.	*a.* Take min. bore diam.
b. Subtract min. groove diam.	*b.* Subtract max. groove diam.
c. Result: twice max. groove depth	*c.* Result: twice min. groove depth
d. Max. groove depth	*d.* Min. groove depth
e. Subtract from min. packing width	*e.* Subtract from max. packing width
f. Result: min. squeeze	*f.* Result: max. squeeze

Example:

a. 1.877	*a.* 1.875
− *b.* 1.501	− *b.* 1.503
c. 0.376	*c.* 0.372
d. 0.188	*d.* 0.186
e. 0.205 − 0.188 = 0.017	*e.* 0.215 − 0.186 = 0.029

3. Gland groove diameter. To obtain gland groove diameter:
   a. Take piston-rod diam.
   b. Subtract negative tolerance
   c. Add twice min. width
   d. Subtract twice min. squeeze
   e. Result: max. gland groove diam.

Example:
a 1.00
−*b* 0.004
0.996
+*c* 0.270
1.266
−*d* 0.024
1.242 max. gland groove diam.

[1] Courtesy of Linear Inc., Philadelphia, Pa.

4. **Determination of range of squeeze**

a. Take max. groove diam.
b. Subtract min. rod diam.
c. Result: twice max. groove depth
d. Max. groove depth
e. Subtract from min. packing width
f. Result: min. squeeze

a. Take min. groove diam.
b. Subtract max. rod diam.
c. Result: twice min. groove depth
d. Min. groove depth
e. Subtract from max. packing width
f. Result: max. squeeze

Example:

a. 1.242
− *b.* 0.996
c. 0.246
d. 0.123
e. 0.135 − 0.123 = 0.012

a. 1.240
− *b.* 0.998
c. 0.242
d. 0.121
e. 0.143 − 0.121 = 0.022

While static O-ring applications are satisfactory for any pressure used in hydraulic work, dynamic applications are not recommended for pressures above 1,500 psi, unless backup rings are used. These are leather rings inserted on both sides of the O ring to prevent extrusion into the piston clearances. Backup rings are advantageous even for lower pressure applications and lengthen the life of the O rings. Backup rings are supplied in sizes shown in Table VII. Rectangular grooves are used with the groove-width at the bottom as given in Table VII, with about 5° taper toward the top, and 1/32-in. bottom radius.

TABLE VII

O-ring width	Backup-ring width	Groove width
1/16	0.062	0.222
3/32	0.062	0.256
1/8	0.062	0.293
3/16	0.094	0.428
1/4	0.125	0.556

Any desired installation of O rings may be computed with the aid of these data. For those who have frequent occasion to determine O-ring installation dimensions, the excellent design handbook published by Linear Inc., Philadelphia, Pa., is recommended.

Finishes recommended for use with O rings are as follows:

Piston rod or cylinder bore: 5 to 15 micro-inches rms

Groove surfaces: 20 micro-inches rms.

O rings must not be used with any but steel cylinders and piston rods. Only hardened and ground or chrome-plated rods are suitable. Wiper or scraper rings, as shown on Fig. 138, are recommended with piston-rod O-ring packings.

In addition to application in rod and piston packings and gasket seals,

O rings have found a multiplicity of uses in the hydraulic industry, as will become apparent in the discussion of valve and control devices. O rings also make an excellent seal for zero or near-zero pressure, where V packings almost always fail. Essential in the success of this type of packing is accurate and smooth finish, hardened rods, and protection from grit, dirt, and abrasives. O rings are not suited for rotating shafts. For this application, homogeneous (Neoprene) V packings are recommended where

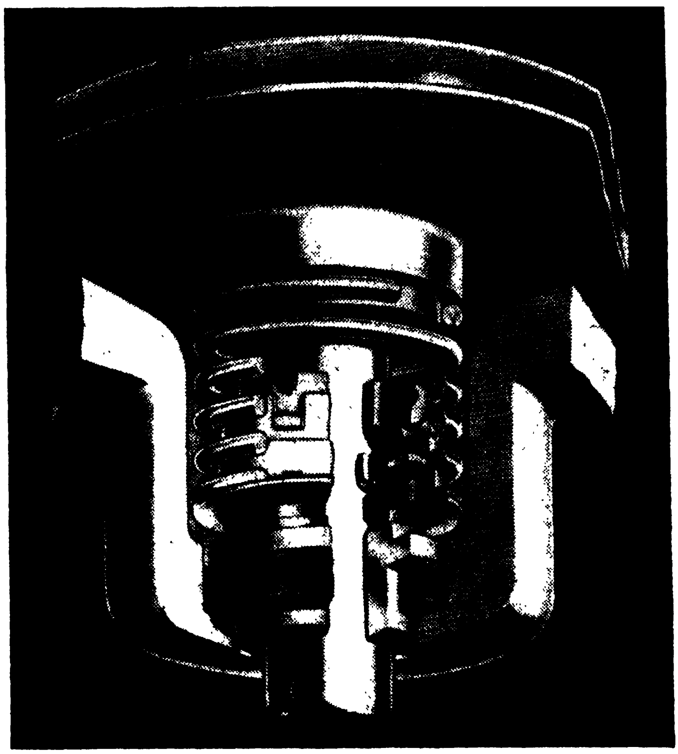


FIG. 139. Crane bellow seal. (*Crane Packing Co., Chicago, Ill.*)

there is an appreciable amount of pressure present. Zero or near-zero pressures may be handled with leather or synthetic seals on either rotating or reciprocating motion. These seals, shaped like a cup packing and with a garter spring or similar expanding device, are mounted in steel housings for ready insertion by press fit. They are available in any desired shaft size and are from 3/8 to 3/4 in. high according to size. The most satisfactory shaft seal for moderate pressure is a mechanical type of seal, such as the John Crane shaft seal. In this device, shown in Fig. 139, the sealing members consist of a carbon disk rotating on a smoothly polished cast-iron seat and a synthetic-rubber bellows, which seals on the shaft and rotates with the shaft. Thus, there is no rotating motion or friction on the shaft proper. A spring, forcing the carbon disk against the seat, produces sufficient pressure to ensure an oiltight seal even at zero pressure. The combination of the spring and flexible synthetic-rubber bellows

allows the carbon disk to advance against the seat as the two faces wear. This type of seal occupies a relatively large space in both length and diameter but is unexcelled where absolutely oiltight seals are required.

Cushioned Cylinders. If hydraulic cylinders, described in the foregoing, are operated at high speeds, special design considerations may be found necessary. Inertia forces in the rapidly moving pistons and parts attached to them may become very large, and means must be provided to permit deceleration and acceleration within reasonable limits. Such means may be provided in the operation of the controlling valves, when

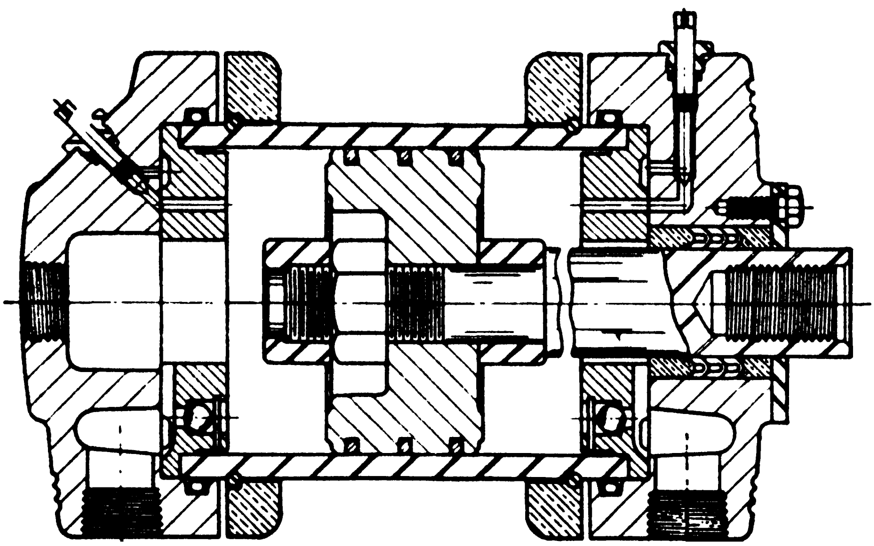


FIG. 140. Cylinder with cushioning devices. (*Gerotor May Corp., Baltimore, Md.*)

pistons are required to stop, start, or change direction in a position intermediate to the extremes of their travel. Oftentimes, however, the hydraulic machines are indexed by letting the pistons come to a positive stop inside of the cylinders. If these pistons operate in excess of 50 ft per min., or if extremely heavy weights are moved, means must be provided to decelerate and accelerate them. To this end, the cylinders are equipped with so-called "cushions."

Figure 140 shows a cylinder equipped with a cushioning device at both ends of the stroke. At the end of the stroke in either direction, a boss on the piston enters a closely fitted counterbore in the head, shutting off the normal escape of fluid displaced by the piston. The escaping fluid is forced to pass through a choke or throttle valve to the exhaust port. The check valve provided in the head permits oil to enter for the return stroke. A still better cushioning effect is obtained by tapering the boss entering the counterbore, so that the throttling effect is produced gradually. V slots will also be found useful for this purpose. Almost all commercially available cylinders may be supplied with these cushioning devices.

CHAPTER IX

THE TRANSMISSION OF HYDRAULIC POWER

1. Piping and Tubing. Hydraulic power, represented by oil flowing under pressure, must be transmitted from the point of generation to the point of application, as electric power is transmitted through wires and conduits. To the ultimate success of a hydraulic installation, the transmission system is as important as the corresponding electrical circuit is to an electrical system. An improperly engineered or carelessly fabricated piping system may ruin an otherwise well-designed hydraulic installation. Oil is conducted to the point of application by piping or tubing. The two terms may be best defined by the statement that the one is originally measured by its inside diameter, while the other is denoted by its outside diameter. A ½-in. pipe originally meant a pipe having a ½-in. ID, while a 1-in. tube is one having a 1-in. OD. The present line-up of pipe sizes is based on this definition, which, however, owing to the introduction of additional weights or wall thicknesses, has lost much of its original meaning. Pipe is available in four weights: standard, now called "Schedule 40"; extra-heavy, now called "Schedule 80"; Schedule 160; and double-extra-heavy. All weights of one size have the same outside diameter, the inside diameter depending on the wall thickness.

Tables showing dimensions, areas, and other specifications for all four weights may be found in any of the standard handbooks. Instead of reproducing them in this text, the author has prepared a table, applicable to oil hydraulic work, which is based on the following empirical facts:

1. Despite several attempts to standardize on a logically conceived system of sizes for hydraulic transmission lines, the use of the so-called "iron pipe size" persists and will hardly be abandoned in the foreseeable future.

2. Only seamless, cold- or hot-rolled pipe should be used for hydraulic lines; butt- and lap-welded pipe is unsatisfactory.

3. Standard-weight Schedule 40 should be entirely eliminated from hydraulic lines owing to lack of mechanical strength, even in low-pressure and suction lines.

4. Double-extra-heavy pipe is not required for oil hydraulic work, as pressures employed rarely exceed 3,000 psi.

5. Use of pipe is not recommended in sizes smaller than ½ in., as tube and tube fittings are much preferable, as will be shown later.

These recommendations result in the line-up of Table I.

TABLE I

Pipe size	OD	Schedule 80, 1,000 psi max.			Schedule 160, 3,000 psi max.		
		Wall thickness	ID	Internal area	Wall thickness	ID	Internal area
½	0.840	0.147	0.546	0.234	Use Schedule 80		
¾	1.050	0.154	0.742	0.433			
1	1.315	0.179	0.957	0.719			
1¼	1.660	0.191	1.278	1.283	0.250	1.160	1.060
1½	1.900	0.200	1.500	1.767	0.281	1.338	1.410
2	2.375	0.218	1.939	2.953	0.343	1.689	2.250
2½	2.875	0.276	2.323	4.238	0.375	2.125	3.550
3	3.500	0.300	2.900	6.605	0.437	2.626	5.410
3½	4.000	0.318	3.364	8.888			
4	4.500	0.337	3.826	11.497	0.531	3.438	9.320

2. Fabrication and Connection. An oil hydraulic system may be made up in any of the following ways, depending on size and service conditions:

1. Screwed fittings and threaded piping, similar to the installation of gas piping. This type of installation should be restricted to pipe sizes not larger than 1¼ in. and pressures not higher than 1,000 psi.

2. Threaded fittings and flared tube. This type of installation is excellent for the smaller sizes of hydraulic systems up to 1 in. and pressures as high as 5,000 psi. Larger installations have been made up to 2 in. and 1,000 psi.

3. Socket-welded fittings and pipe with welded flanged connections. This system should always be used on pressures over 1,000 psi and for sizes above 1¼ in.

Simultaneous employment of methods 3 and 1 will result in an amateurish-looking installation and should be avoided.

On hydraulic systems, regardless of size or pressure, only forged fittings, T's, elbows, or unions should be used, with no cast fittings of any kind. One of the finest lines of fittings for hydraulic work is manufactured by the Watson Stillman Company, Roselle, N.J. Recommendations given in the following are based on their catalogued standards.

Method 1. For method 1, Schedule 80 pipe may be used, as listed in Table I, together with 2,000-lb Watson Stillman screwed fittings. T's, elbows, 45's, crosses, and unions are available in all standard pipe sizes

listed above. Hexagon and flush reducing bushings, as well as couplings, plugs, and caps may also be supplied. Connection to units of the hydraulic system are generally made by screwing the pipe directly into the tapped openings; sometimes units are supplied with tapped flanges for this purpose. It is not customary to bend pipe in this method of installation, owing to the difficulty of tightening any leaks that may develop. All threads should be made to ASA-B2 standards. Great care must be taken in assembly to keep inside of pipes free of shavings, cutting compound, pipe dope, etc. The author has found that tight joints may be made by tinning the threaded end of the pipes.

Method 2. In method 2, fittings are employed having a male or female pipe thread on one end, while the other end is adapted to receive a

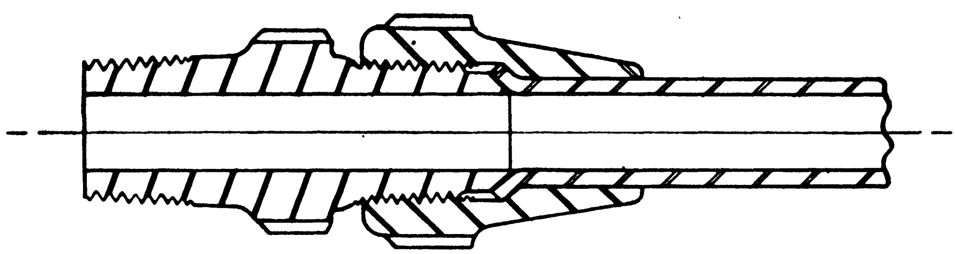


FIG. 141. Tube connector. (*V. L. Graf Co., Detroit, Mich.*)

union nut to secure tubing to the fitting, making an oiltight, quickly disconnectible union connection, which may be made and broken repeatedly without danger of leakage. Numerous shapes of fittings are available in a number of different makes, consisting of straight connectors, elbows, T's, and crosses with male or female pipe connections, and also elbows, T's and crosses with all tubing connections. All manufacturers' fittings are coded as to size and type and may be selected from their catalogues. Tubing to fit is supplied and denoted by outside diameter and wall thickness. Figure 141 shows section of a straight connector with the tube assembled and secured by flaring.

While most manufacturers carry numerous combinations of pipe and tube sizes, the author has found it advisable to standardize on one tube size for each pipe size, so that the inside area of the tube is approximately equal to the area of the corresponding Schedule 80 pipe. This results in a line-up applicable to any standard make of fittings, as given in Table II. Tubing used in connection with these fittings should be dead-soft annealed SAE 1010, drawn to close tolerances to permit close fit in tube union-nuts.

Tube is fabricated by bending with a minimum of fittings. Suitable bending machinery, either hand- or power-operated, is available for this purpose. Tubing should not be bent to a radius of less than three diameters. All the tubing listed in Table II may be bent cold on hand benders without filling or other preparations. Care should be taken to

flare tubes properly. Flares must be of correct sizes and free from cracks. Special flaring tools may be purchased to facilitate this operation.

Method 3. This type of piping installation may be used for two pressure ranges: Schedule 80 for 1,000 psi in. in all sizes from ½ to 4 in. and higher, and Schedule 160 from 1¼ in. and up for 3,000 psi. Schedule

TABLE II

Pipe size	Tube OD	Wall	Tube ID	Area	Pressure, psi*
⅛	¼	0.035	0.180	0.025	3,000
¼	⅜	0.060	0.250	0.050	3,000
⅜	½	0.070	0.350	0.100	3,000
½	⅝	0.095	0.435	0.150	3,000
¾	⅞	0.110	0.630	0.300	2,500
1	1⅛	0.120	0.875	0.600	2,000

* 5: 1 safety factor.

80 is good also for 3,000 psi for 1 in. and smaller. Socket-welding fittings are available for both weights. For Schedule 80, Watson Stillman 3,000-lb fittings are recommended, and for Schedule 160, Watson Stillman 4,000-lb fittings should be used. Elbows, T's, 45's, crosses, and couplings are available. Unions up to 2 in. in both Hex and two-bolt flange may be supplied. The piping system is made up by slipping the ends of the pipe into the socket of the fitting and welding. Figure 142 shows the dimensions of the fillet weld recommended for Watson Stillman fittings. The weld is approximately triangular in section. This type of weld, when properly made, will give satisfactory service. The author prefers counterboring the fittings and using both groove and fillet weld, as shown in Fig. 143. A J groove is most satisfactory, as it permits penetration of the arc to the bottom of the groove.

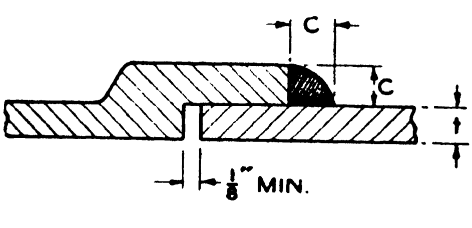


FIG. 142. Fillet weld for pipe fitting. (*Watson Stillman Co., Roselle, N.J.*)

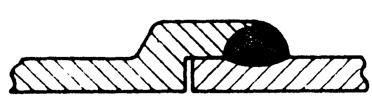


FIG. 143. Groove and fillet weld.

When it is desired to connect pipes of different sizes together, fittings may be machined specially for reduced sizes, or socket-welding reducers may be supplied. Pipe is bent to avoid use of fittings where possible. While smaller sizes may be handled cold on bending machines, large pipes

are generally bent hot and filled with sand before bending. A minimum radius of five pipe diameters is recommended.

The method of connecting the piping to the different units of the hydraulic system consists of socket welding the pipe into a flange, which

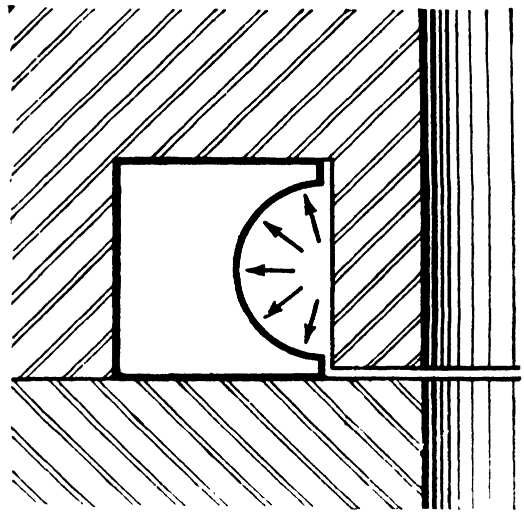


FIG. 144. The Vickerseal. (*Vickers Inc., Detroit, Mich.*)

in turn is bolted to the unit with the aid of suitable gaskets. No generally adopted standard is available for these flanges, and a number of manufacturers have developed standards of their own. The flanges are

TABLE III

Pipe size	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*
1/2	3 3/4	2 1/2	1 1/2	7/8	23/32	0.870 0.868	1 5/16	1.120 1.118	1/8	1/8	3/4	1 1/2	1 1/2	1/2	3/4	1/4	3	1 7/8	3 3/8	1/4	3/8
3/4	3 3/4	2 1/2	1 3/4	1 1/16	3/4	0.870 0.868	1 1/2	1.120 1.118	1/8	1/8	7/8	1 3/4	1 3/4	1/2	3/4	1/4	3	2 1/8	3 5/8	1/4	5/8
1	4 1/2	3	2	1 11/32	15/16	1.120 1.118	1 7/8	1.370 1.368	1/8	1/4	1	2	2 1/4	3/4	7/8	1/4	3 1/2	2 1/2	4 1/4	5/16	3/4
1 1/4	5 1/2	3 3/4	2 1/2	1 11/16	1 1/16	1.245 1.243	2 5/16	1.495 1.493	3/16	1/4	1 1/4	2 1/2	2 3/4	7/8	1	1/4	4 1/2	2 7/8	5 1/8	3/8	5/8
1 1/2	6	4	2 3/4	1 15/16	1 5/16	1.495 1.493	2 9/16	1.745 1.743	3/16	1/4	1 3/8	2 3/4	3	1	1 1/8	1/4	5	3 1/4	5 3/4	3/8	3/4
2	7 1/2	5	3 1/2	2 13/32	1 5/8	1.995 1.993	3 1/8	2.245 2.243	3/16	1/4	1 3/4	3 1/2	3 3/4	1 1/4	1 3/8	3/8	6	4	7	1/2	1
2 1/2	9	6	4	2 29/32	2	2.370 2.368	3 3/4	2.620 2.618	3/16	3/8	2	4	4 1/2	1 1/2	1 5/8	1/2	7	4 5/8	8 1/8	1/2	1 1/8
3	9 1/2	6 1/2	5	3 17/32	2 1/2	2.870 2.868	4 1/2	3.245 2.243	1/4	1/2	2 1/2	5	5 1/2	1 3/4	1 5/8	1/2	7 1/2	5 1/8	8 7/8	5/8	1 3/8
3 1/2	11	8	5 1/2	4 1/32	3	3.495 3.493	5	3.995 3.993	1/4	1/2	2 3/4	5 1/2	6 1/4	2	1 7/8	1/2	9	6	10 1/2	5/8	1 1/2
4	12 1/2	9	6	4 19/32	3 1/2	3.870 3.868	5 1/2	4.245 4.243	1/4	1/2	3	6	7 1/4	2 1/4	2	1/2	10	6 5/8	11 5/8	5/8	1 5/8

generally of the four-bolt type of heavy section to minimize deflection and bored for welding the pipe similarly to the fittings. Gaskets may be of the confined compression type (Fig. 132) or O ring (Fig. 133.). A type of seal that has come into use recently is the self-tightening Vickerseal (Fig. 144). This seal is available in a number of sizes to conform to standard pipe sizes. Bolts or cap screws should be calculated with conservative stresses to allow not only for the hydraulic pressure loads, but also for stresses imposed by misalignment, relative movement of the components of the hydraulic system, etc.

Flanges with both face and lateral outlets or a combination of both may be used advantageously. Figure 145 shows dimensions of standard face-outlet and side-outlet flanges made by Hydro-Power Inc., Mount Gilead, Ohio. These flanges in matched pairs may be used as unions, for coupling lengths of piping together, in sizes larger than 2 in.

Greatest care must be taken to remove sand, scale, and dirt from the interior of the pipe after fabricating. In making the installation, the pipe must not be forced into place; forcing may distort hydraulic valves and other components of the system.

Correct dimensioning of the piping system is of greatest importance for the efficiency of the entire hydraulic system. Computation of losses and recommendations for velocities to be allowed have been covered in Chap. VI. Particular care must be taken to compute the actual rate of flow in a pipe to arrive at the correct diameter. The sample computation carried out in Chap. VI illustrates this process.

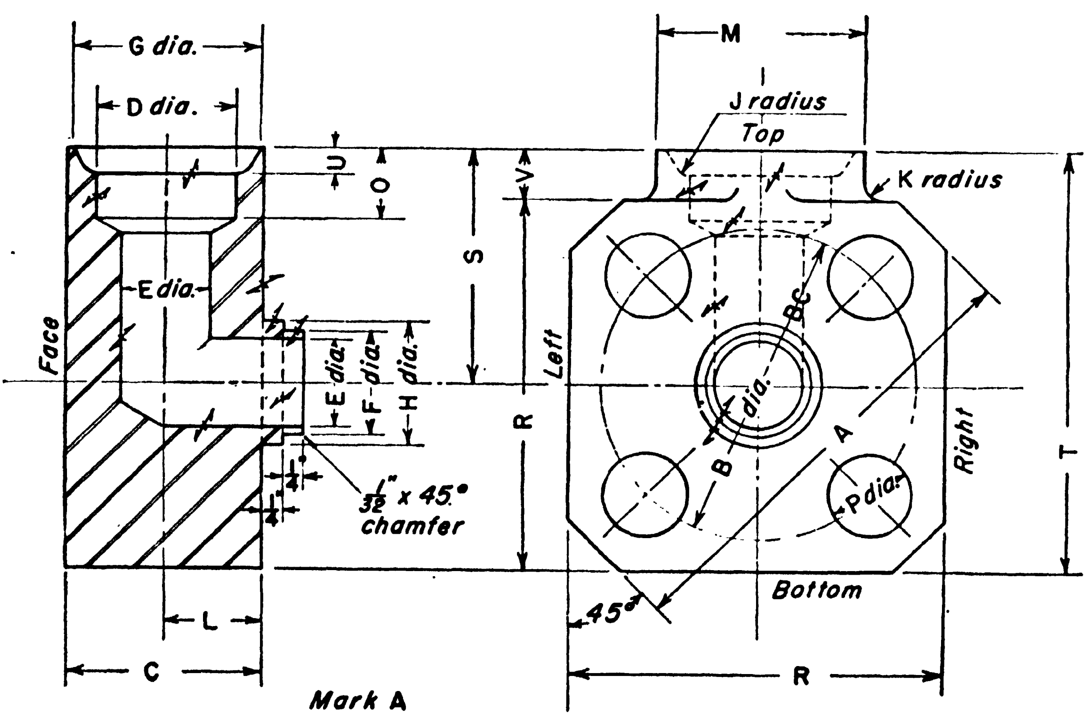


FIG. 145. Standard flanges. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

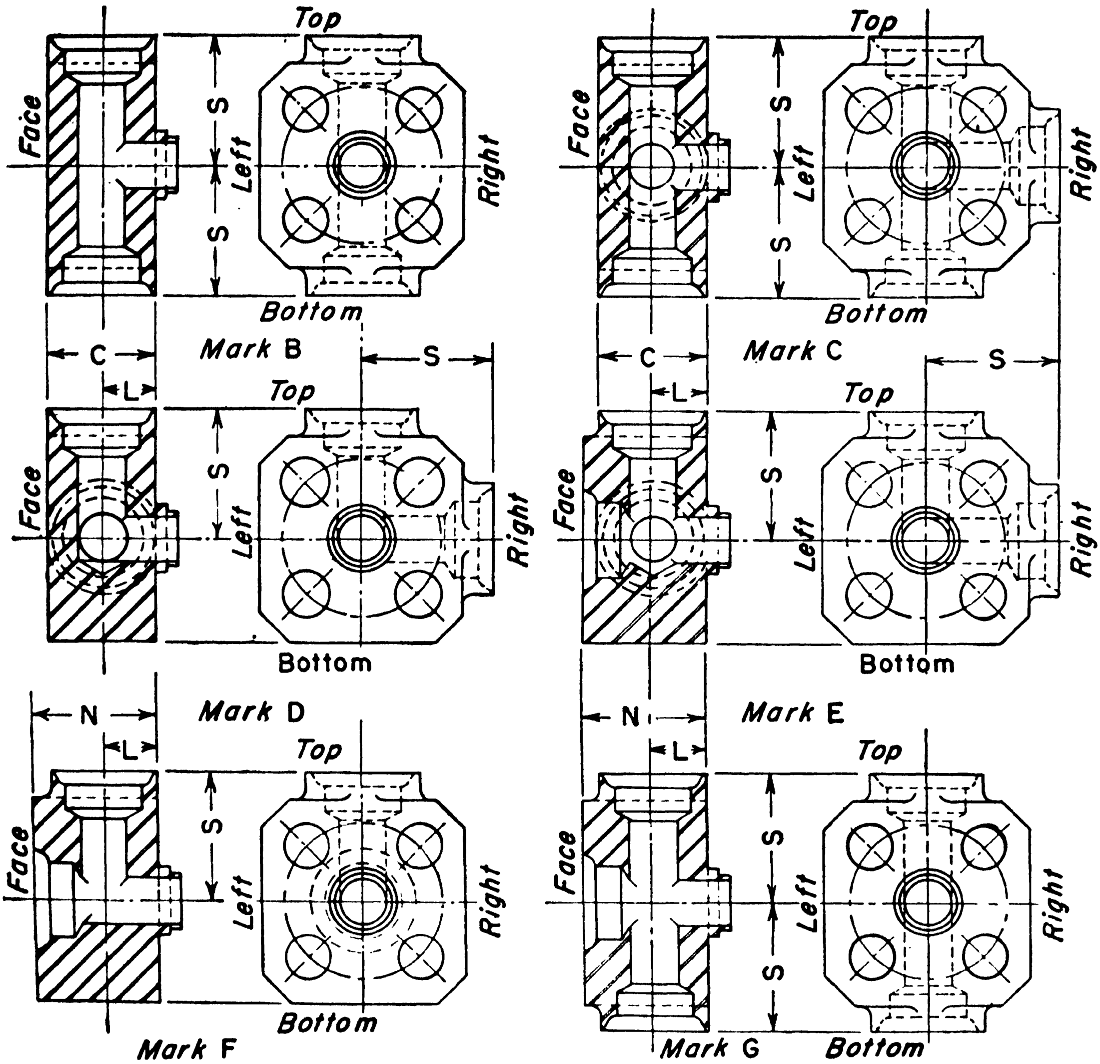


FIG. 145—(*Continued*).

TABLE IV

Pipe size	A	B	C	D	E	F	G	H	J	P	Q	R	U	W	X	Y	Z	O	AA
1/2	3 3/4	2 1/2	1	7/8	3/4	0.870	1 5/16	1.120	1/8	3/4	1/4	3	1/4	1/2	0.875	1.125	3/4	1/2	5/8
						0.868		1.118							0.877	1.127			
3/4	3 3/4	2 1/2	1	1 1/16	3/4	0.870	1 1/2	1.120	1/8	3/4	1/4	3	1/4	3/4	0.875	1.125	3/4	1/2	5/8
						0.868		1.118							0.877	1.127			
1	4 1/2	3	1 1/4	1 11/32	15/16	1.120	1 7/8	1.370	1/8	7/8	1/4	3 1/2	5/16	1	1.125	1.375	7/8	3/4	3/4
						1.118		1.368							1.127	1.377			
1 1/4	5 1/2	3 3/4	1 3/4	1 11/16	1 1/16	1.245	2 5/16	1.495	3/16	1	1/4	4 1/2	3/8	1 1/4	1.250	1.500	1 1/4	7/8	7/8
						1.243		1.493							1.252	1.502			
1 1/2	6	4	2	1 15/16	1 5/16	1.495	2 9/16	1.745	3/16	1 1/8	1/4	5	3/8	1 1/2	1.500	1.750	1 1/4	1	1
						1.493		1.743							1.502	1.752			
2	7 1/2	5	2	2 13/32	1 5/8	1.995	3 1/8	2.245	3/16	1 3/8	3/8	6	1/2	2	2.000	2.250	1 3/8	1 1/4	1 1/4
						1.993		2.243							2.002	2.252			
2 1/2	9	6	2 1/2	2 29/32	2	2.370	3 3/4	2.620	3/16	1 5/8	1/2	7	1/2	...	2.375	2.625	...	1 1/2	1 1/2
						2.368		2.618							2.377	2.627			
3	9 1/2	6 1/2	3 1/2	2 17/32	2 1/2	2.870	4 1/2	3.245	1/4	1 5/8	1/2	7 1/2	5/8	...	2.875	3.250	...	1 3/4	1 1/2
						2.868		3.243							2.877	3.252			
3 1/2	11	8	3 3/4	4 1/32	3	3.495	5	3.995	1/4	1 7/8	1/2	9	5/8	...	3.500	4.000	...	2	1 3/4
						3.493		3.993							3.502	4.002			
4	12 1/2	9	4	4 19/32	3 1/2	3.870	5 1/2	4.245	1/4	2	1/2	10	5/8	...	3.875	4.250	...	2 1/4	1 7/8
						3.868		4.243							3.877	4.252			

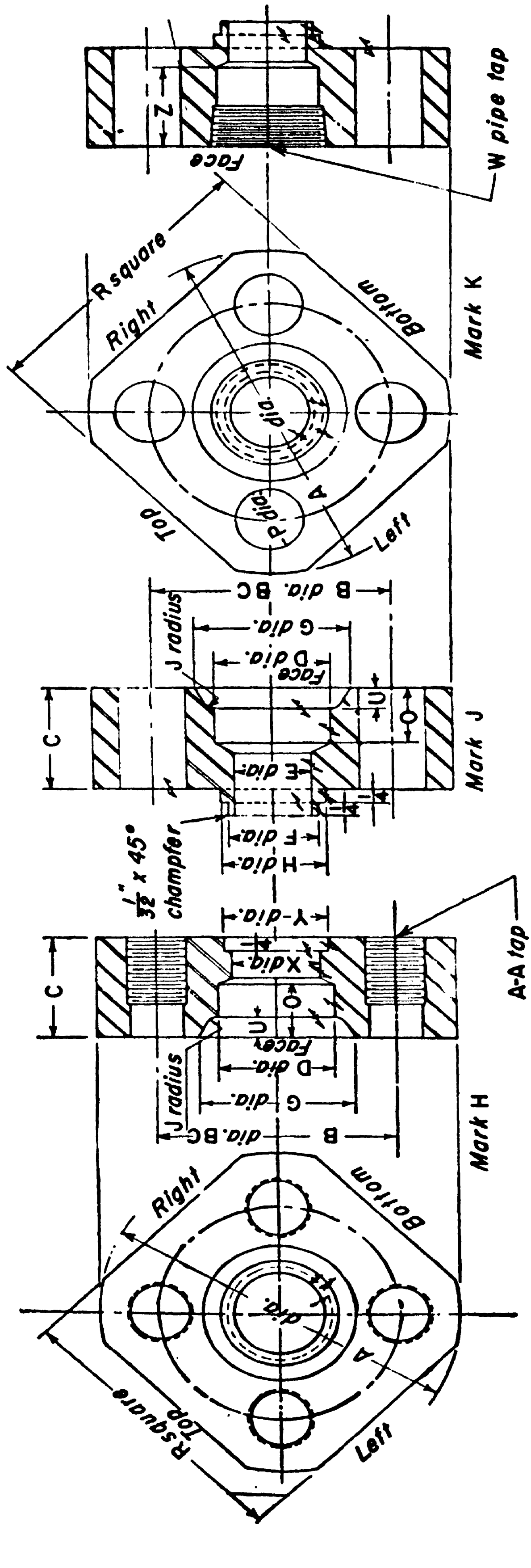


FIG. 145—(*Continued*).

CHAPTER X

THE CONTROL OF OIL HYDRAULIC POWER

1. GENERAL TYPES OF HYDRAULIC CONTROLS

Hydraulic power generated by pumps and supplied to hydraulic motors must be controlled in order to achieve the functions desired of the power devices. The controls may be divided into the following functional divisions:

Pressure Controls. These devices limit the pressure applied by the pump to the circuit, or may determine pressures existing in any part of the circuit. They also serve to unload the pumps during part of the machine cycle or determine the pressure at which oil is admitted to or exhausted from parts of the circuit. According to any of these requirements, we distinguish between relief and safety valves, reducing valves, unloading and admission, or sequence, valves, and back-pressure valves. Any of these devices either control or are actuated by hydraulic pressure.

Directional Controls. These devices direct the flow of fluid to the parts of the system where its application is desired. This may be done manually or automatically by mechanical, hydraulic, or electric means. Among these controls we find operating valves of the two-way, three-way, and four-way types, which may be manually, hydraulically, mechanically, or electrically actuated. There are also pilot valves for operating other valves and control functions, such as check, stop, and shutoff valves, which again may be manually or automatically actuated. Multiple-unit control valves, which combine several control functions in one body, are available.

Volume Controls. The volume delivered by constant- and variable-delivery pumps may be controlled in a number of ways. We have simple throttle valves, flow controls and flow dividers, and special remotely controlled metering devices for variable-delivery-pump circuits.

Combination Controls. Two or more of the control means for pressure direction and volume may be combined in self-contained units grouped in valve blocks or control panels. The entire functional control of a hydraulically actuated machine may be centered in a single hydraulic control panel. Standardized control panels and units are supplied by a number of manufacturers of hydraulic equipment.

Storage and Intensification. It is of interest to note that the storage and intensification of hydraulic power is quite old in the art of hydraulics

and antedates the introduction of oil hydraulics. For some time after this, the scheme of storing and intensifying hydraulic power fell into discard, owing to the development of flexible oil-pressure hydraulic generators of high efficiency. Very recently these methods have revived because of the invention of air-ballasted, synthetic-rubber, bladder-type accumulators, as originally developed for aircraft use, and new designs of continuous intensifiers, as exemplified by Racine and Hydro-Power.

2. PRESSURE CONTROLS

Relief Valves. The simplest pressure control is the spring-loaded relief valve. In its most elementary form, the valve, shown in Fig. 146, consists of a body with inlet connection in which is mounted a plunger that covers the outlet opening and is held down by a spring with adjustable tension. Pressure entering the intake opening acts upon the exposed area of the plunger, and when the total force of the hydraulic pressure upon this area exceeds the setting of the spring, the plunger will lift and permit the oil to relieve itself into the exhaust opening. This elementary form of valve, while frequently used owing to its low cost and simplicity, has certain disadvantages that may be overcome by modifications to be discussed later.

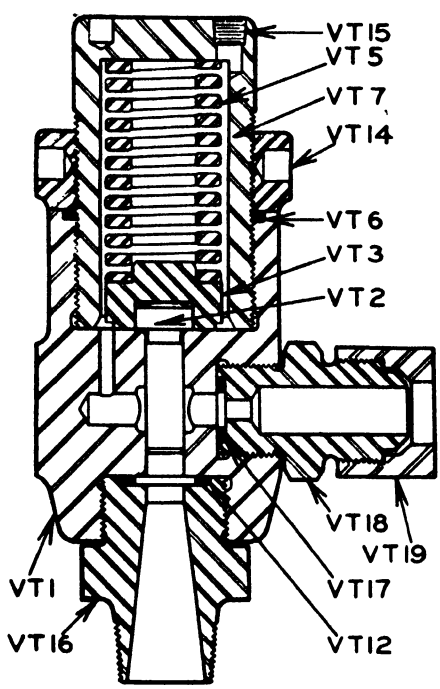


FIG. 146. Spring-loaded plunger-type relief valve. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

In high-pressure applications, the force applied to a piston of sufficient area to permit relief of the volume of oil at reasonable velocity is very great. This leads to heavy springs, and with these heavy forces opposing each other, there is a tendency to chatter, present in all valves of this type, which often are extremely noisy. The cause for this condition is that at the instant the valve lifts, there is a sudden pressure drop under the plunger, which causes it to reseat suddenly, in turn causing a sudden pressure rise; this again lifts the valve, and the action continues to repeat, resulting in violent chatter. When testing these valves, it will be found that a valve set to open at a given pressure with a hand pump, will maintain a much higher pressure when subsequently operated by a power-driven pump of larger capacity. In fact, the pressure rise is a function of the pump capacity or the amount of fluid attempting to pass the valve.

This is not, as often erroneously assumed, due to the spring gradient, but is caused by the conversion of pressure into velocity at the opening line of the valve. General design information will be given in the following concerning salient features of this type of valve.

Velocities through the bore of the valves generally should not exceed 50 ft per sec. With permissible discharge line speeds of 10 to 12 ft per sec, the area of a relief-valve plunger could therefore be from one fifth to one-fourth of its pipe-size area, or roughly the plunger diameter will be one-half the nominal pipe diameter. This will lead to correct dimensions and reasonable differential pressures (about 10 per cent of setting) for

Table I

Wire size	Max. working stress, psi	Max. stress variation in operation, psi
$\frac{1}{8}$ or less	70,000	40,000
Over $\frac{1}{8}$ to $\frac{3}{16}$	70,000	30,000
Over $\frac{3}{16}$ to $\frac{3}{8}$	60,000	25,000
Over $\frac{3}{8}$ to $\frac{5}{8}$	50,000	20,000

pressures from 1,000 to 3,000 psi. High-pressure safety valves, designed for overload protection and not for continuous operation, may be operated at velocities as high as 100 ft per sec, resulting in considerably higher differentials. Plungers are hardened and ground high-carbon steel (SAE 52100). Sometimes, but not always, hardened sleeves or seats are used, especially in cases of frequent operation. Cast- or forged-steel bodies are used, sometimes meehanite or high-tensile iron. The distance that the plunger extends into the bore, known as "lap," should be about equal to one-third the plunger diameter. Fits between plungers and bores may be made in accordance with recommendations in Table I, Chap. V, although some manufacturers use lapped fits to avoid all preliminary leakage loss. Spring gradients may be based upon about 20 per cent increase in spring tension due to lifting the lap distance. Differential pressure thus created has not been found additive to dynamic differential. Pressure-adjustment ranges, maximum to minimum, generally are not greater than 4:1 for this type of valve. Wider ranges require the use of several springs.

Springs in these valves are an important item. Alloy-steel or music wire is recommended to reduce their physical sizes. Stresses should be held within limits dictated by good practice. The working stresses given in Table I are recommended for springs made from Swedish wire or alloy steel, if long fatigue life is essential. Where fatigue is not an issue, as in safety valves, which are used only occasionally, or where extreme space

limitations are present and replacement at reasonable intervals is not objectionable, stresses up to 50 per cent in excess of those listed may be used. Wahl correction factors should be used in all spring calculations. Best designs have D/d ratios of about 8. Design information on springs may be found in any engineering handbook and need not be gone into here. The annoying chatter of this type of relief valve may be relieved by throttling or choking the outlet openings, as shown in Fig. 146, where

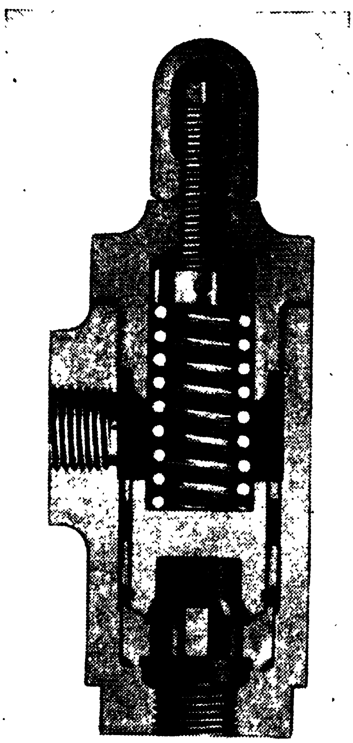


FIG. 147. Oilgear foot-valve-type relief valve. (*The Oilgear Co., Milwaukee, Wis.*)

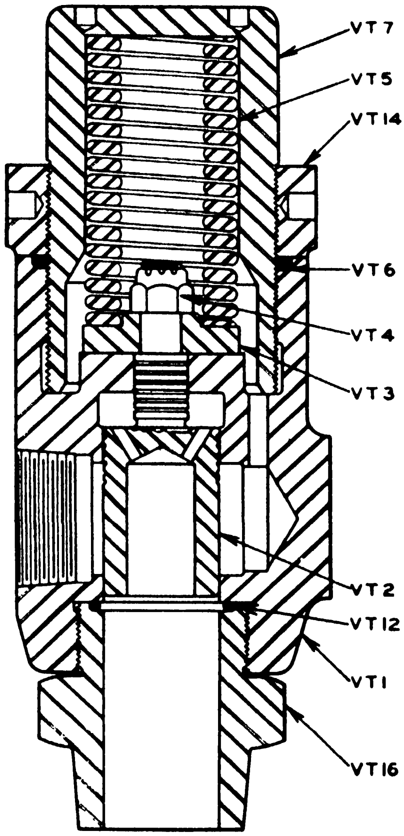


FIG. 148. Differential relief valve. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

an orifice union is used at the outlet. This again has a tendency to raise the differential pressure, but will produce reasonably quiet operation. Oil dashpots have also been used successfully for the same purpose. Figure 147 shows the Oilgear foot-valve type of relief valve, which features a dashpot type of seat that cushions and quiets the operation.

Simple plunger relief valves for low pressures may be made in most any size required. In high-pressure valves of this type over 1 in. in size, spring proportions become very large, and a number of design modifications have been made to cope with this difficulty. The most generally used is the differential relief valve. This type of valve has made its appearance in various modifications, all based on the principle that a

small lifting area is provided, opposed by a relatively light spring that actuates the valve at the preset pressures and opens a large relief area to relieve a larger volume. Generally, the lifting area should not be less than one-fourth of the relieving area, but even with this stipulation, great saving in size and bulk may be accomplished.

Figure 148 shows a differential relief valve made by Hydro-Power Inc. Pressure entering at the bottom of the large plunger passes through openings into the space above this plunger and neutralizes part of the upward thrust so that the remaining load to be resisted by the spring is that of the pressure acting on the small stem extension. A large area is presented to the escaping fluid. Partial balance of pressures on the valve plunger tends to quiet the operation of the valve.

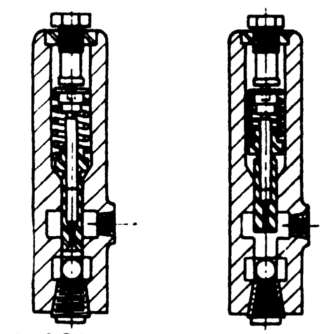


Fig. 149. Type QS relief valve. (*The Oilgear Co., Milwaukee, Wis.*)

A reduction in spring size and bulk is also accomplished in the Type QS Oilgear relief valve, shown in Fig. 149, owing to employment of a stationary plunger that partly balances the hydraulic load on the relieving plunger.

The Gerotor May relief valve, shown in Fig. 150, has an independent

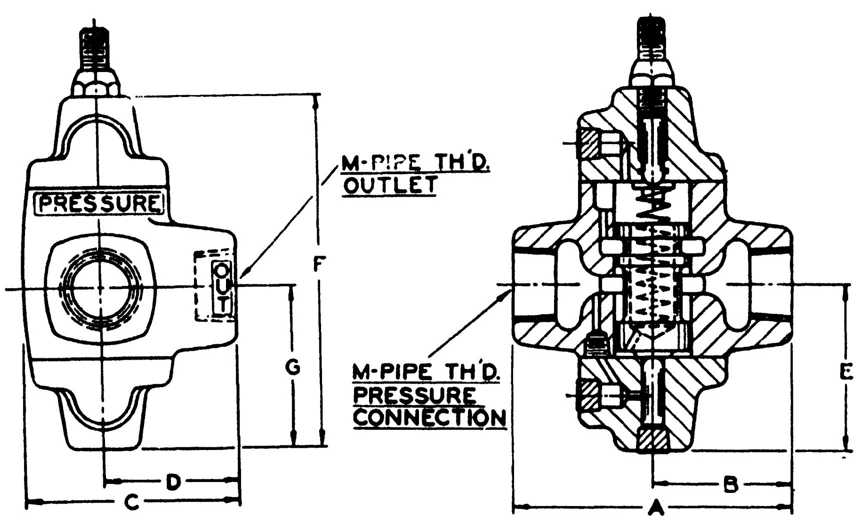


Fig. 150. Pilot-operated relief valve. (*Gerotor May Corp., Baltimore, Md.*)

lifting plunger, operating against adjustable spring tension to lift a balanced valve piston that relieves the pressure fluid into the exhaust.

Differential relief valves may be made in almost any desired capacity and size. Pipe connections may be flanged or pipe threaded according to requirements. In keeping with recommendations of Chap. IX, valves over $1\frac{1}{4}$ in. should be made with flanged connections. Differ-

ential relief valves may be made with damping arrangements to eliminate all chatter. This may be done by admitting pressure to the lifting area through a check valve to ensure quick lifting in case of sudden pressure rise, and letting pressure escape from the lifting area through a choke valve to dampen the spring action in reseating it.

The Vickers Balanced Relief Valve. This valve consists in principle of a piston having equal areas on both sides. Oil flows straight through the valve underneath the piston, which has a lower extension, forming a valve

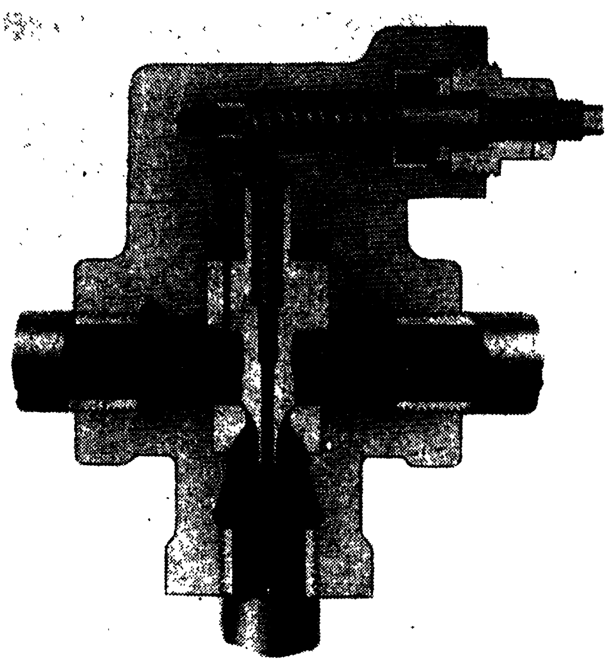


FIG. 151. The principle of the Vickers balanced relief valve.

plunger to close the exhaust opening. The oil may pass through a small passage to the upper side of the piston, which has an extension equal in area to the lower extension. The upper extension opens into a chamber connected with the exhaust through a bore in the piston. A small spring-loaded pilot relief valve is mounted in such a way that pressure in the chamber above the piston may operate the valve and escape into the exhaust. A light spring holds the main valve-piston assembly toward the closed position. In operation, hydraulic pressure passing through the restriction in the valve piston keeps the valve in perfect hydrostatic balance, until the pilot relief valve opens, permitting the fluid above the piston to escape. Owing to the hydrostatic balance of the piston, the slightest pressure drop causes movement of the piston to open the exhaust. A valve of this type has almost zero differential, remaining fully closed almost up to the point of maximum setting and then relieving the full volume without increase in pressure. There is no chatter in this type of

valve, because of the balanced-piston construction, and the operation is smooth and quiet at all pressures. An exterior view of the Vickers valve is shown in Fig. 152.

A very important feature of the Vickers relief valve is the possibility of dropping the pressure in the system, governed by the relief valve, by remote control. This action, termed "venting" by Vickers engineers, consists in opening a by-pass around the pilot relief valve, by either a small manually actuated or a cam-controlled valve. When this is done,

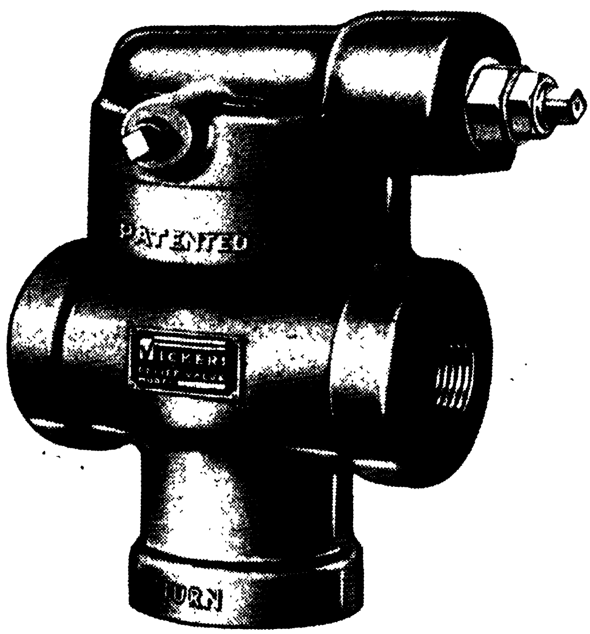


FIG. 152. Vickers relief valve. (*Vickers Inc., Detroit, Mich.*)

the balanced piston is immediately unbalanced, and the valve will open wide, permitting escape of pump volume at practically zero pressure. Thus a hydraulically operated machine may be unloaded at the end of a cycle to idle at low power consumption. The valve may also be remotely actuated by auxiliary pilot valves, which may be controlled in a desired manner by manually or automatically actuated shutoff valves. A multiplicity of functions is thus made possible with this valve. Pressure in the hydraulic system may be lowered or raised at desired cycle intervals automatically or manually, and the system may be entirely unloaded, if so desired, in a similar manner. Figure 153 shows the schemes of operation described above.

Back-pressure Valves. Back-pressure valves, also called "foot" or "balancing" valves, are a modification of simple relief valves. Their purpose is to prevent gravitational descent of a hydraulic ram or piston that operates vertically, or to create a back pressure in the exhaust end of

a double-acting cylinder to operate a hydraulic ram "locked" under hydraulic pressure, thus preventing sudden lunging in case of sudden relief of load, as, for instance, in the breaking through of a drill in a hydraulically operated drill press. To this end, the valve consists of a combination of a regular relief valve with a check valve that permits flow in reverse direction for the return or retraction stroke. Direct spring-loaded valves of single or differential plunger type are used for this

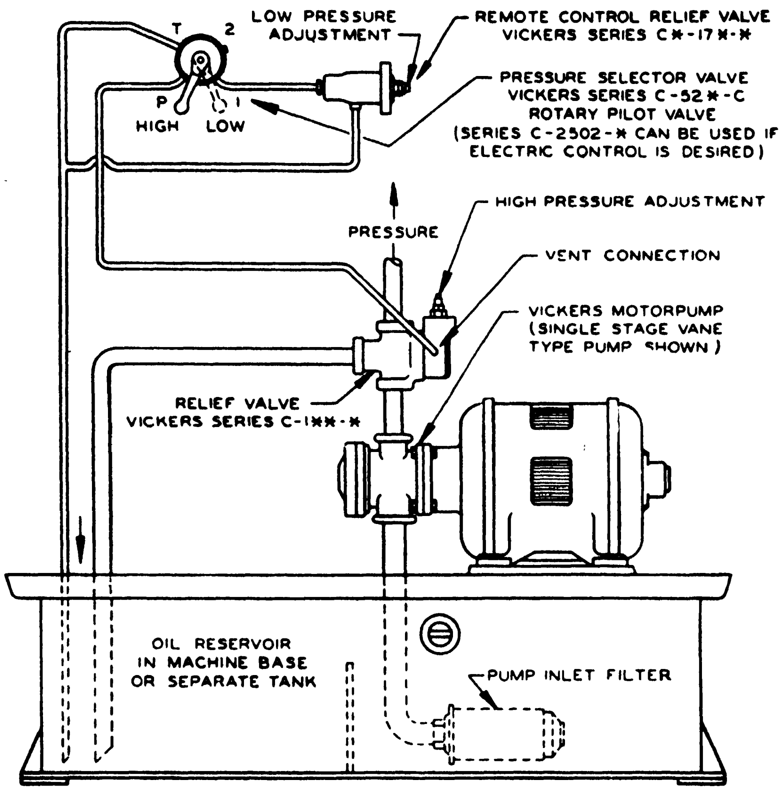


FIG. 153. Application of venting and remote control of Vickers relief valve. (*Vickers Inc., Detroit, Mich.*)

purpose, generally with built-in check valves. The rather unique design in Fig. 154 shows the Oilgear foot valve. In all these valves, the entire valve body, including spring housing and adjustment screw, are subjected to full return-system pressure and must be designed accordingly.

A design of back-pressure valve that serves as its own check valve is shown in Fig. 155. If the lift area is about one-fourth of the release area, then the pressure to open the valve on the retraction stroke is only one-fourth of the setting of the forward resistance, which is generally negligible. Separate exhaust drain of the spring chamber must be provided to permit this valve to open on return flow.

Sequence Valves. A further modification of the spring-loaded relief valve is the sequence valve. Sequence valves are pressure-operated valves that control the sequence of flow to various portions of an oil

circuit. Since these valves are often confused with the back-pressure or foot valves described above, a thorough distinction must be made, and to this end the definition of their operation by Vickers Inc. will be quoted. "Operation is such that flow at pressure inlet is directed to pressure outlet until a pressure equal to the valve setting is built up. Full pressure is then available at both outlet connections, and the valve acts as a T in the piping, as long as pressure is maintained above the valve setting." From this it becomes apparent that a valve such as shown in Fig. 154 is not a sequence valve in the sense of this definition, as it will act as a reducing valve when the pressure in the outlet begins to build up, and full pressure cannot be reached in the outlet connection. The outlet pressure must be prevented from acting on the valve plunger so that it may build up to the full value of the inlet pressure. To this end, the connection between outlet chamber and spring chamber (see Figs. 146 and 148) is eliminated, and a separate connection to drain is made from the spring chamber. The reader will recognize by examining Fig. 150 that provisions are made on the Gerotor relief valve to do this. All that is necessary is to plug the connection between outlet chamber and spring chamber and remove the plug in the spring-chamber vent and connect it

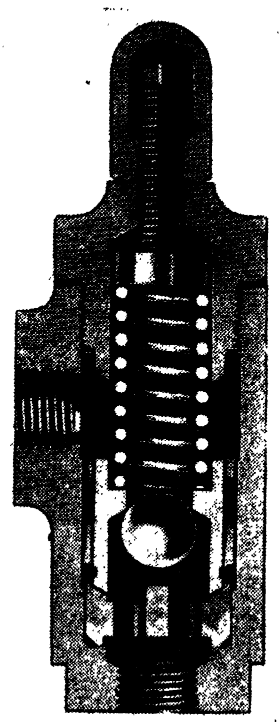


FIG. 154. Foot valve. (*The Oilgear Co., Milwaukee, Wis.*)

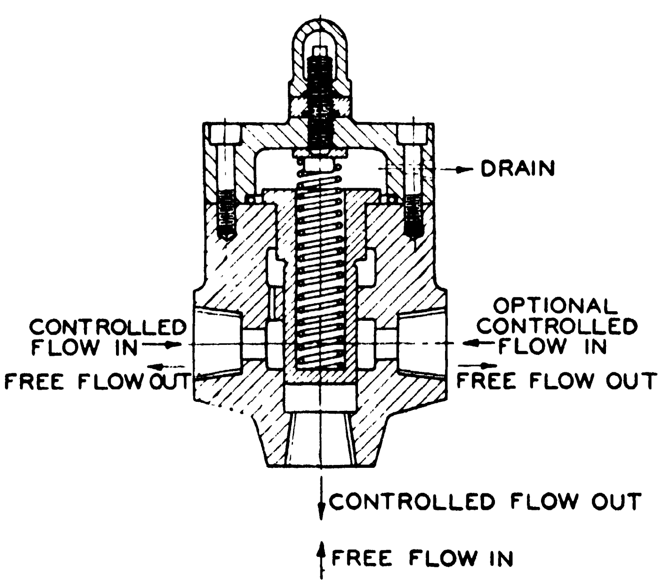


FIG. 155. Reverse-flow back-pressure valve.

to the drain. In the valve shown in Fig. 155, the outlet pressure acts on the plunger in opposite direction to that shown in Fig. 154, assisting the opening of the valve, once the spring setting has been reached and oil has started to flow into the outlet. This means the valve will not reseat until the pressure has fallen far below the original setting. Obviously this design would not make a good sequence valve.

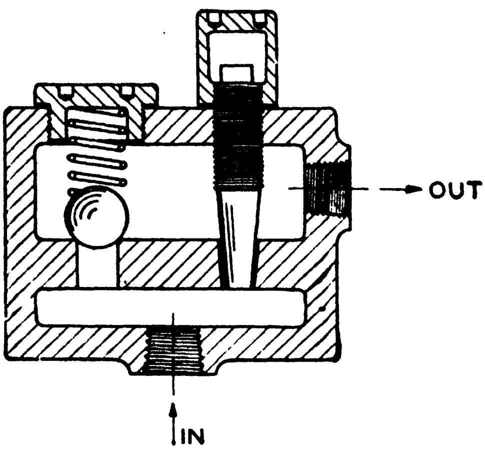


Fig. 156. Damping arrangement for relief and sequence valves.

Sequence valves may be made with built-in check valves to permit free reverse flow. The author has found that this type of valve is particularly given to chattering or hammering at the point of shift-over, and cushioning or damping devices should be provided in the design. This may be done by regulating the passage leading to the lifting area (see Fig. 150) by a choke screw or choke pin. An arrangement that permits "pin-point" opening of the valve and controlled closing, is shown in Fig. 156. This device, which may be built into the valve or connected separately by pilot lines, permits free flow through the ball check valve into the lifting or actuating area and controlled reseating by means of an adjustable tapered choke pin. This device, applicable to all types of pilot or auxiliary area-operated relief or sequence valves, will positively eliminate chatter and hammering upon opening.

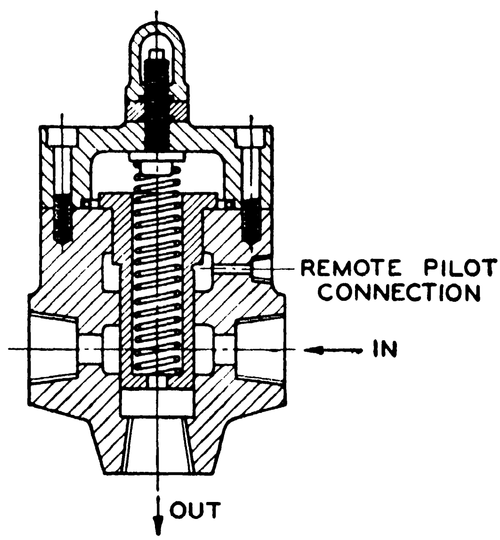


Fig. 157. Unloading valve.

Unloading Valves. Valves heretofore described serve to maintain pressure at a predetermined setting on an oil-pressure circuit or parts of the circuit, while oil is discharged through them at the set pressure. A valve described in the following will permit the pump to build up to an adjustable setting and then unload to by-pass at practically zero pressure, as long as pressure setting is maintained by a remote source of pressure. This so-called "unloading" valve is generally used in combination pumping units, having high- and low-pressure pumps, where the low-pressure pump is fully unloaded at its pressure limit and the high-pressure pump

carries on to a higher pressure setting. Unloading valves may also be used to advantage to relieve large volumes of oil from the large area of

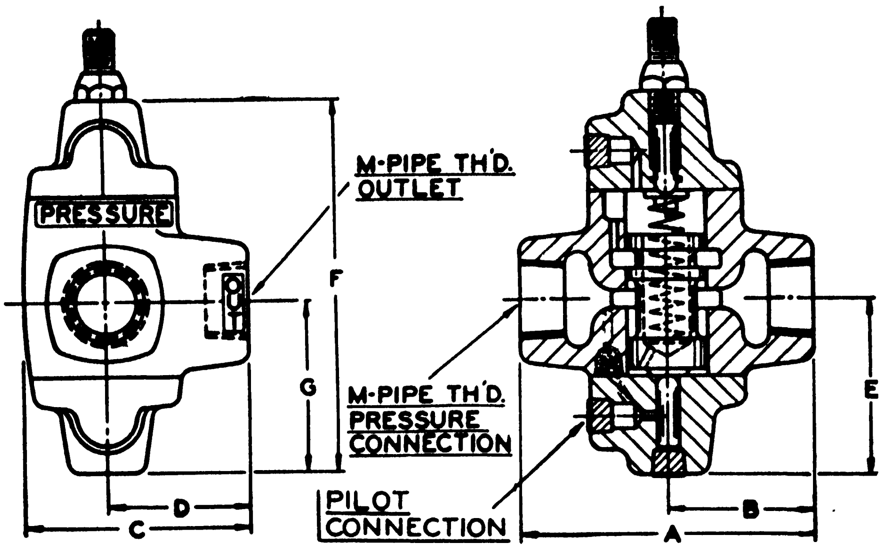


Fig. 158. Unloading valve. *(Gerotor May Corp., Baltimore, Md.)*

double-acting cylinders, when pressure is applied to the small area for the retraction stroke. Since full release of pressure and full flow with minimum resistance are desired, areas of unloading valves should be made from 75 to 100 per cent of their rated pipe area. A commonly used design of unloading valve is shown in Fig. 157. The Gerotor May unloading valve is shown in Fig. 158.

Vickers Inc. makes a complete line of counterbalance, sequence, and unloading valves of the so-called "Hydrocushion" type. Hydrocushion is their trade name for hydraulic dashpots to prevent chatter and hammering. An exterior view of their valve is shown in Fig. 159.

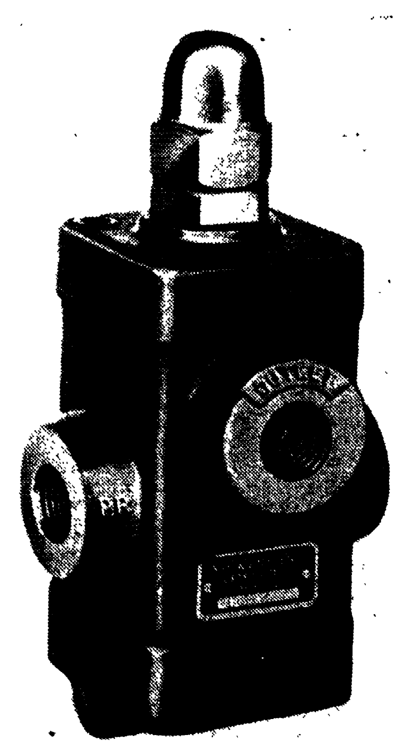


Fig. 159. Hydrocushion valve. *(Vickers Inc., Detroit, Mich.)*

A valve unit used in connection with Vickers combination high- and low-pressure pumps is shown in Fig. 160. Low-pressure oil flows from opening *A* through passage *E* and check valve *C* into passage *D*, where it is joined by high-pressure oil from inlet *B*. At a given preset pressure, the unloading valve opens, actuated by pressure from passage *G*, which connects *E* to *H* and unloads the low-pressure pump. The high-pressure pump continues to deliver volume through passage

D, seating check valve *C*. Finally relief valve *F* opens at the preset maximum limit and relieves the high-pressure pump.

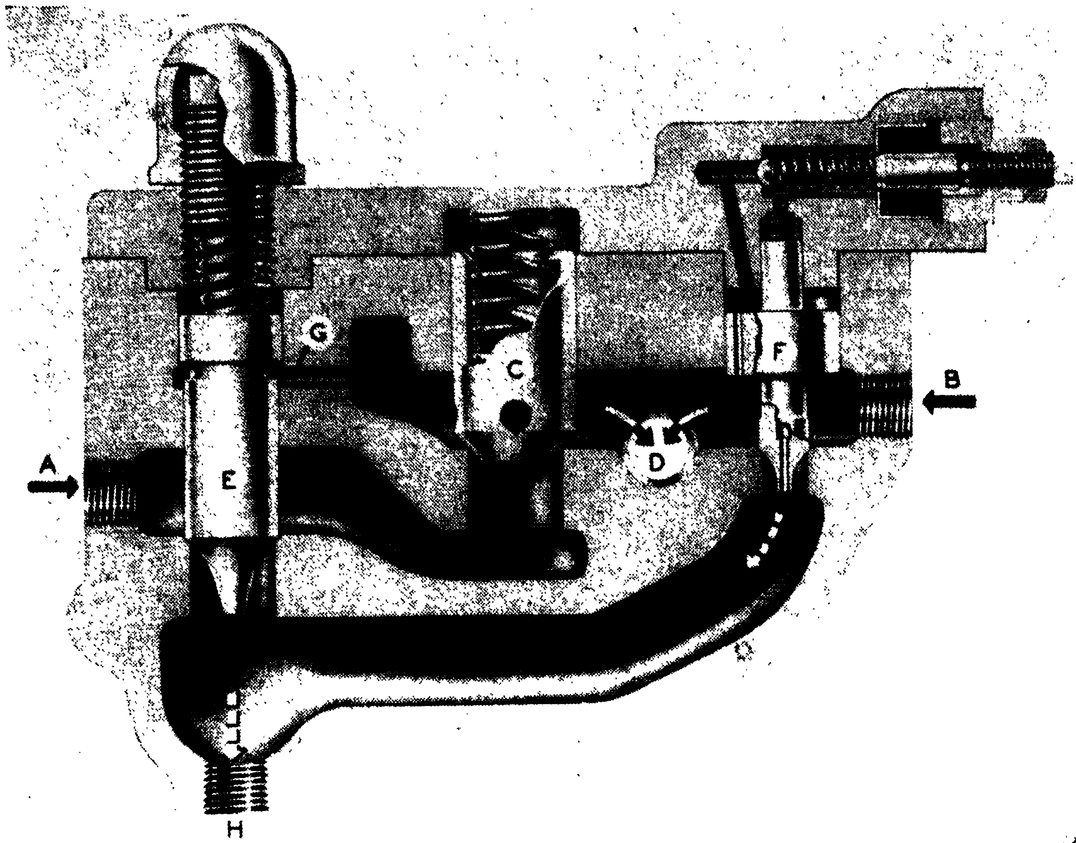


FIG. 160. Vickers combination valve, schematic arrangement.

The Load-dividing Valve. A specialized form of relief valve is the load-dividing valve, used where two pumps are operated in series and it is desired to have each pump carry one-half the load. This valve is a form of differential relief valve with a light spring load. The operating principle is shown in Fig. 161. The first stage, or low-pressure pump, has a capacity slightly in excess of the second or high-pressure unit and delivers oil to the intake of the high-pressure unit. The load-dividing valve is placed so that the discharge line of the low-pressure unit is connected to the larger of the two areas, tending to open the valve and relieve into the exhaust.

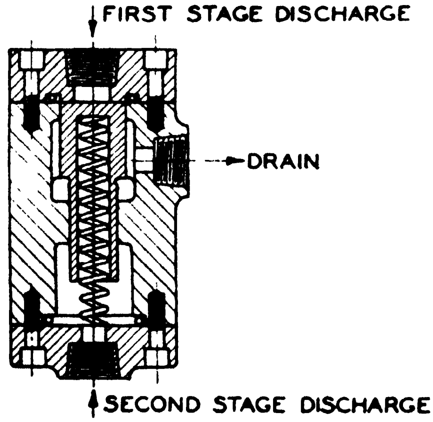


FIG. 161. Load-dividing valve.

The smaller of the two areas is connected to the discharge of the high-pressure unit and opposes this tendency. With the areas made in a

ratio of 2:1, any outlet pressure of the high-pressure stage, determined by load requirements or relief-valve setting, will automatically produce

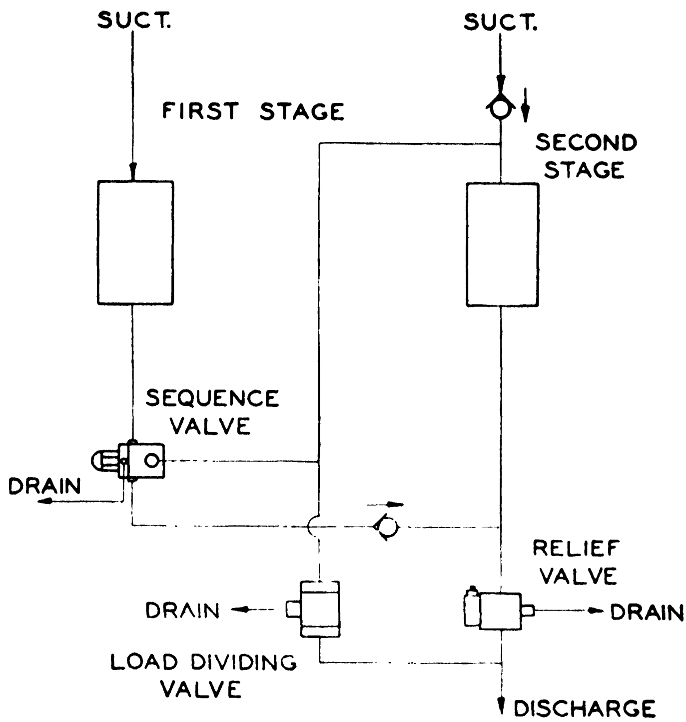


FIG. 162. Series-parallel operation of two-stage unit.

one-half this pressure in the low-pressure stage. This type of valve is used in Vickers two-stage vane pumps. Series-parallel units may be controlled by sequence valves in the parallel connection, which open at a preset pressure in the discharge line of the first stage and cause discharge into the intake of the second stage. A diagram of this scheme of operation is shown in Fig. 162. Both pumps operate in parallel, until a pressure slightly below one-half the maximum is reached. At this pressure the sequence valve will open and admit outlet flow from the first stage into the inlet of the second stage. Pump will then continue to operate in series at one-half the output and twice the pressure.

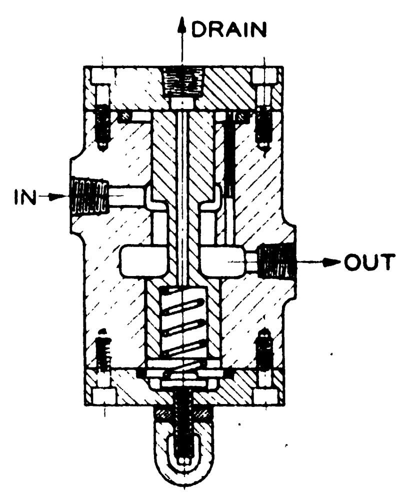


FIG. 163. Reducing valve with constant reduced pressure.

Reducing Valves. The purpose of a reducing valve is to maintain a reduced pressure in a portion of a hydraulic system. Two basic types of reducing valves may be distinguished: those which maintain a fixed reduced pressure in a portion

of the circuit, regardless of the magnitude of pressure in the rest of the circuit, and those maintaining a fixed reduction of pressure, resulting in reduced pressure varying with system pressure.

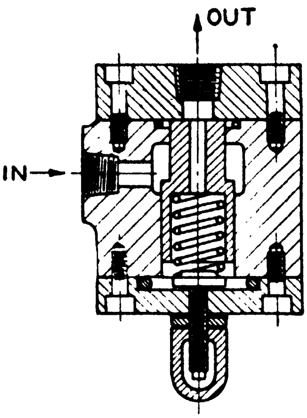


Fig. 164. Reducing valve with constant reduction.

A spring-loaded valve of the first variety is shown in Fig. 163. Fluid enters at the intake and passes through the opening in the valve to the outlet, until the pressure on the differential plunger builds up to the setting of the spring, when the valve will close and limit the pressure in the outlet to that determined by the spring setting. Both ends of the differential plunger are vented to drain. To prevent gradual build-up of pressure in the outlet, a leak-off is provided, consisting of a metering pin in a small bore connecting the outlet to drain.

The second variant is illustrated in Fig. 164. This valve is a modification of a regular differential relief valve. Pressure, determined by the spring setting, will open the valve and permit escape of fluid into the outlet. Outlet pressure prevails on both ends of the differential piston. This pressure will always equal the system pressure less a fixed reduction determined by spring setting. Again a bleed-off should be provided in the outlet to prevent equalization of pressure due to leak from inlet to outlet.

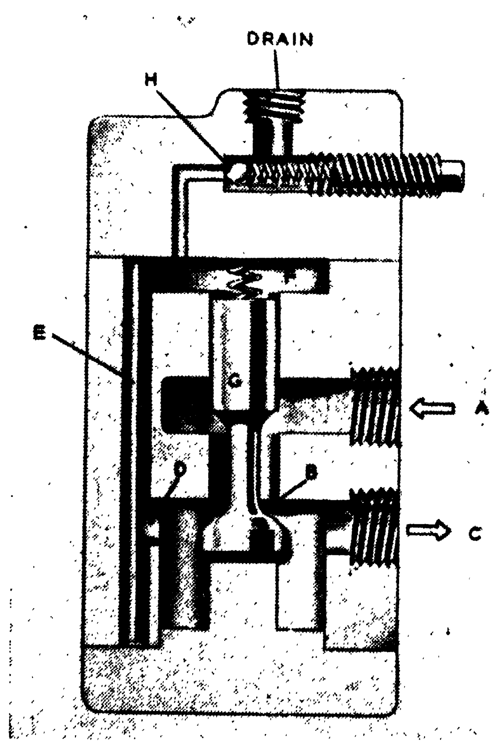


Fig. 165. Principle of the Vickers reducing valve.

The Vickers reducing valve, illustrated in Fig. 165, operates to produce a fixed reduced pressure in the outlet like the valve in Fig. 163, utilizing the Vickers balanced-piston principle instead of the customary spring-loaded construction. Oil enters the valve at the intake *A* and passes through the open valve into outlet passage *C*. Outlet passage *C* is connected to pilot-relief-valve chamber *F* through a restricted passage *E*. As pressure builds up in outlet *C*, it will bleed through passage *E* and balance valve plunger *G*, which cannot close until the setting of the pilot valve is reached. When this pilot valve opens,

plunger G becomes unbalanced and closes, preventing further flow to the outlet chamber C.

3. DIRECTIONAL CONTROLS

Directional controls serve to direct the flow of hydraulic power to the point where its application is desired. Most commonly used for this purpose are piston-type valves, which for general oil hydraulic usage have almost entirely superseded all other types. In its simplest form, a piston-type control valve consists of a hardened and ground valve piston, closely fitted in a housing and provided with piston heads to direct supply

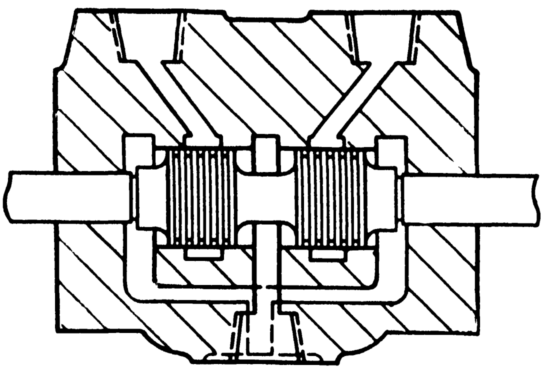


Fig. 166. Blocked-center operating valve. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

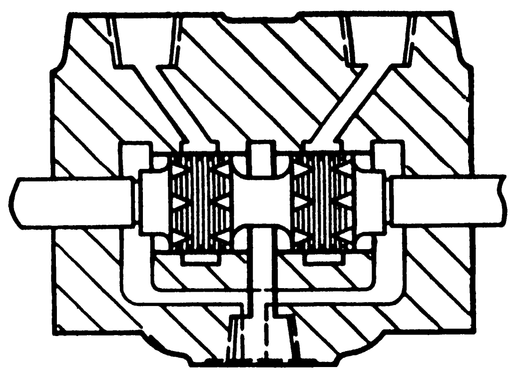


Fig. 167. Open-center operating valve. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

and exhaust to openings from which oil is conveyed by piping attached to the valve casing.

Figure 166 shows a schematic diagram of a simple four-way-type piston operating valve in its neutral or mid-position. In this particular design of valve, pressure is blocked in the neutral or mid-position of the valve and may be utilized to operate other devices. Another design provides for so-called "open-center" position, where a by-pass is provided for the pump in mid-position and both cylinder connections are connected to the exhaust. This design is diagrammatically illustrated in Fig. 167. V-shaped slots are provided to make connection between inlet and cylinder annulus and between cylinder annulus and exhaust. It may readily be seen how oil will pass from intake to the left-hand cylinder connection and from the right-hand cylinder connection to the exhaust, when the valve is shifted to the left, and vice versa, when it is shifted to the right. For smoothest operation, an open-center valve is recommended, especially when made with the V-slot arrangement shown in Fig. 167. When passing the neutral position, there is no sudden blocking of pressure with subsequent shock, as there is in the blocked-center valve.

In addition to the two types of piston design, a number of other combinations are possible with the two-head type of piston, and these are illustrated in the chart (Fig. 168).

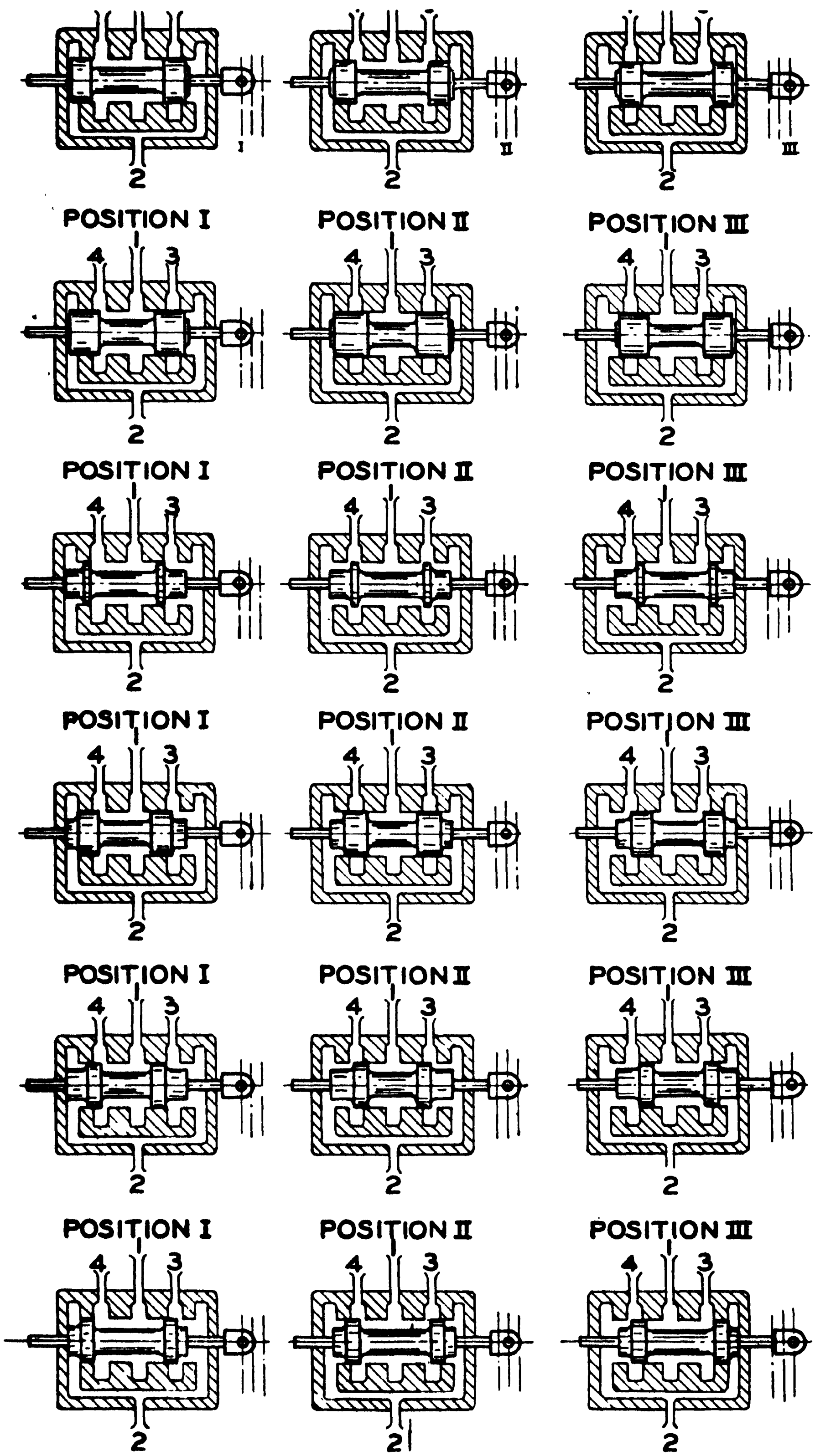


Fig. 168. Plunger functions of three- and four-way control valves. (*The Oilgear Co., Milwaukee, Wis.*)

Valves may be made with spring attachments, either for spring return to the neutral position or with the spring force holding the valve in either end position, the so-called "spring-offset" type. A valve of the latter type, as made by Hydro-Power Inc., is shown in Fig. 169.

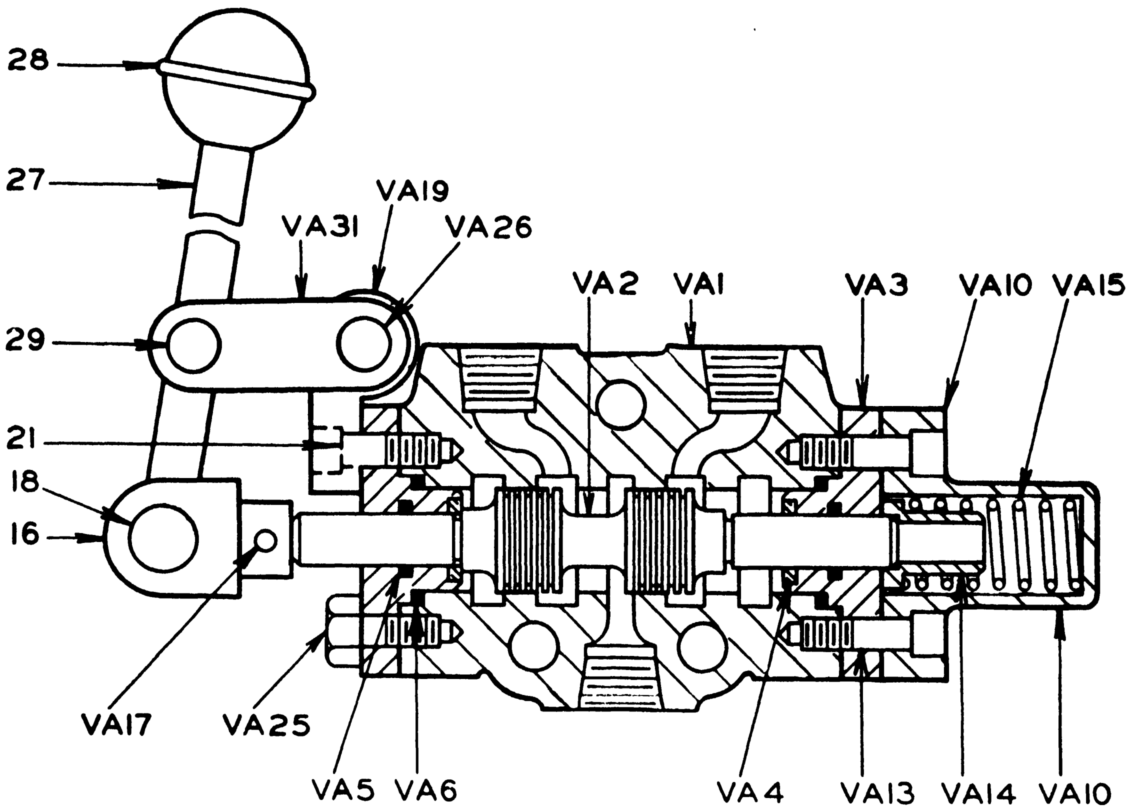


Fig. 169. Spring-offset valve. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

Design Information for Piston Valves. Valve bodies are made of meehanite or alloy iron of close grain and high tensile strength, valve pistons of SAE 2315 or equivalent, carburized and hardened. O-ring packings on valve stems are becoming more and more prevalent.

Fits should be in conformity with information in Table I, Chap. V, if made interchangeably. Lapped fits are sometimes resorted to if minimum-leakage requirements are stringent. For dimensions of O-ring grooves and fits, reference may be made to Tables V and VI, Sec. 3, Chap. VIII. Internal areas should be from 75 to 100 per cent of corresponding Schedule 80 or 160 pipe. In open-center valves, as shown in Fig. 167, velocities of 35 ft per sec may be allowed through the V passages with valve in neutral position. Flanged connections should be used on valves larger than $1\frac{1}{4}$ in. Valve pistons should be grooved to minimize locking effect under pressure. One-sixteenth-inch U-shaped grooves about $\frac{3}{16}$ centers are very satisfactory. V grooves are not recommended owing to the possibility of width variations arising from grinding after hardening.

Valves operating at high pressures must have balanced exhaust (double tail rod) to avoid sudden jerks when large oil volumes are released into unbalanced exhaust. Even with greatest care in design and machining, some locking effect will be present, owing to unbalance on circumferential area. This effect becomes aggravated when the valve is held under pressure for longer periods. Practical experience has shown that spring-actuated valves should have spring pressures of at least 20 lb per in. of piston diameter to break valve piston loose at the end of a pressure-holding period.

TABLE II

Valve size	Piston diam.	Rod diam.	Port slot width	Min. lap	Min. port opening	Total stroke
1/4	5/8	1/2	1/4	1/8	1/16	3/8
1/2	7/8	5/8	3/8	1/8	1/8	1/2
3/4	1	5/8	3/8	1/8	1/8	1/2
1	1 1/4	3/4	1/2	3/16	3/16	3/4
1 1/2	1 1/2	7/8	5/8	3/16	1/4	7/8
2	2	1 1/4	3/4	1/4	5/16	1 1/8

The physical dimensions given in Table II have been found satisfactory and may serve as design guide. Figures are absolute minima for satisfactory operation and may be increased, if desired. The cap screws holding the valve bonnets at the ends should be calculated for full system

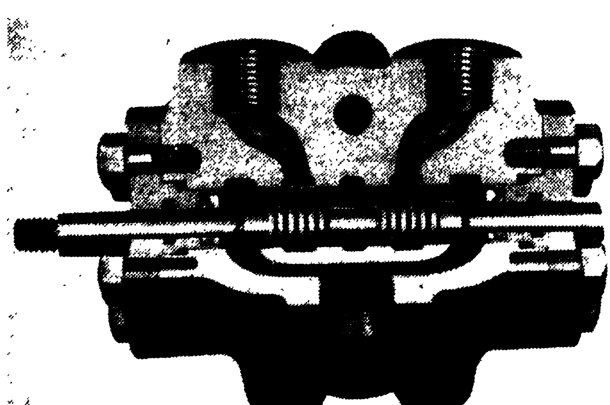


FIG. 170. Operating valve. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

pressure, to care for sudden release of pent-up volumes. Directly manually actuated valves are not generally made larger than 1½ in.

The designer should use great care in development of the transition from the port slot to the circular openings to avoid bottlenecks. The patternmaker should be provided with a number of sections taken horizontally through the ports, so that a core box may be made that provides

constant cross section throughout the travel from pipe opening to piston space.

A great variety of valves are commercially available, made by practically all manufacturers of hydraulic components. Vickers makes a complete line of valves from ¼ to 1 in. in all styles and combinations. The units are small and compact in size, the 1-in. valve measuring about 13 in. over-all length including hand lever and spring housing, and the

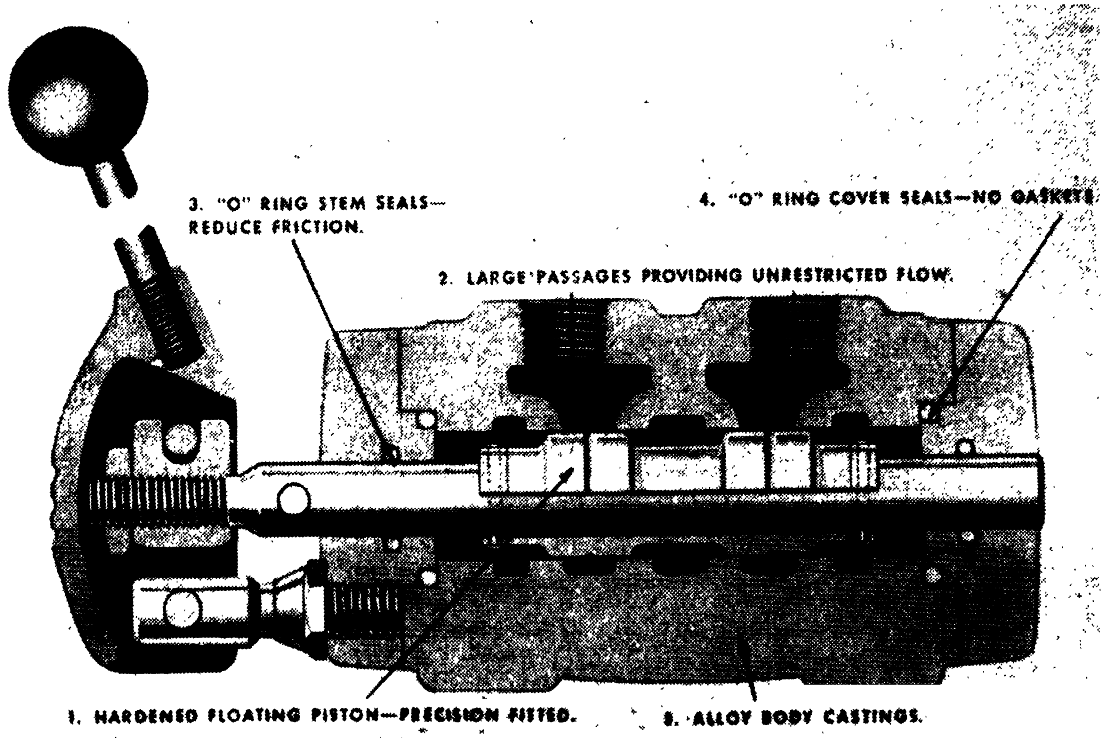


FIG. 171. Operating valve. (*Gerotor May Corp., Baltimore, Md.*)

½-in. about 10 in. Hydro-Power Inc. provides a similar line-up. Section of Hydro-Power valve is shown in Fig. 170.

Figure 171 shows the Gerotor May valve. This is a very excellent design featuring a floating piston, balanced tail rod, and O-ring packing. Figure 172 shows the valve of the Racine Tool and Mfg. Co. in Racine, Wis. This valve has an inserted sleeve with drilled ports and is shown with centering spring. Exterior view of an Oilgear valve with flanged pipe connection is shown in Fig. 173.

Valves described in the foregoing are two-piston-head valves, which permit a number of functions, as shown diagrammatically in Fig. 168. More complex functions may be achieved by increasing the number of piston heads on a valve stem and adding port slots. Three or more piston heads may be used and almost any desired characteristic obtained. Thus five-way, six-way, and other combinations may be created.

One of the most popular applications of a multiple-head valve is the center-by-pass combination with both cylinder openings blocked. This function cannot be achieved with a two-head piston valve. Principle of this valve is shown in Fig. 174. The valve, as illustrated, has a central or

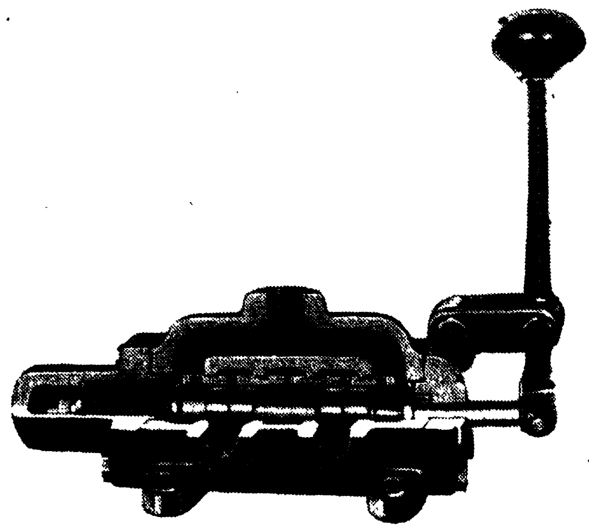


FIG. 172. Operating valve. (*Racine Tool and Mfg. Co., Racine, Wis.*)

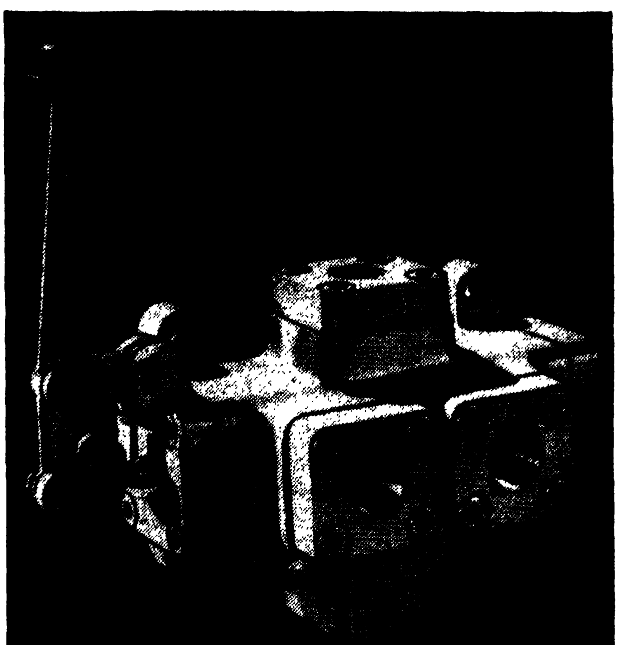


FIG. 173. Four-way operating valve. (*The Oilgear Co., Milwaukee, Wis.*)

distributing portion and an intake and outlet portion flanged to the central part. In neutral position, oil passes freely from the intake to the outlet, while both cylinder ports are blocked. If the valve piston is shifted to either end position, pressure is supplied to the corresponding

cylinder port with the central by-pass blocked, while the opposite cylinder port is exhausted. One of the interesting features of this valve is that any desired number of control sections may be installed in tandem, thus forming a "sandwich" of control valves, each serving a controlled unit. With all valves in neutral, the pump will by-pass freely through

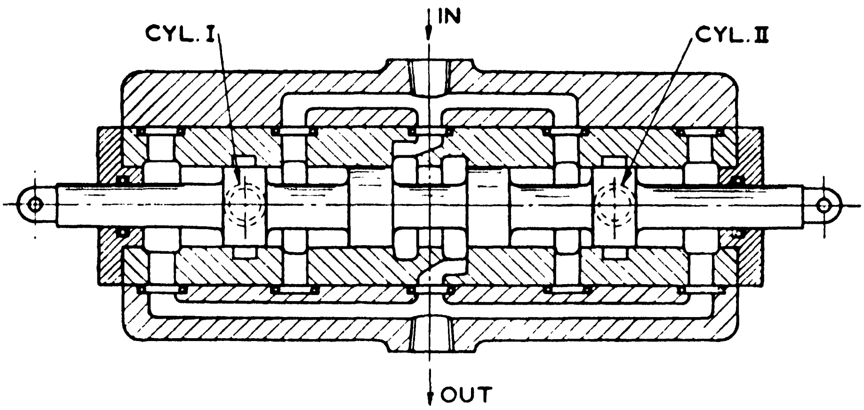


FIG. 174. Center-by-pass control valve.

all port openings in center into the exhaust opening, while all cylinder ports are blocked. Shifting any one of the control valves will actuate and supply pressure to the corresponding cylinder unit. Generally valves are made with centering springs, and the intake flange has a built-in relief valve. The valve made by Vickers Inc. is shown in Fig. 175.

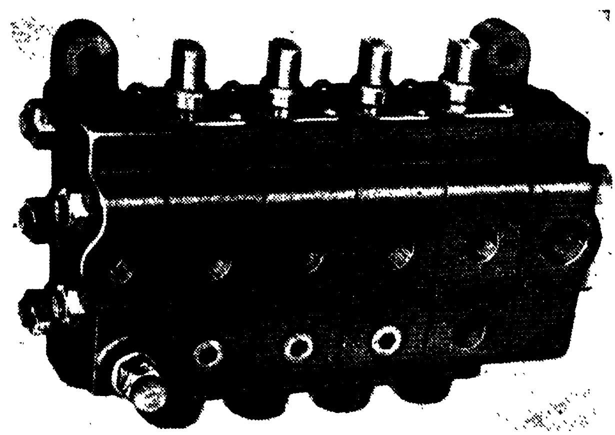


FIG. 175. Multiple tandem control valve. (*Vickers Inc., Detroit, Mich.*)

Innumerable combinations of functions may be produced by variations in the design of the simple piston valve, and the most complicated hydraulic machine may be controlled with the modifications of this simple device. A tandem type of change-over valve, made by the Oilgear Co., is shown in Fig. 176.

Pilot Valves and Pilot-operated Valves. The valves described in the preceding pages are direct manually operated. Balanced construction

and accurate workmanship ensures easy operation within the customarily used range of working pressures. The larger the valves are, the harder they are to operate, and on valves over 1½ in. in size the effort may become excessive. To overcome this difficulty, these valves may be equipped with power pistons and actuated by small auxiliary valves, called "pilot" valves. Operation by pilot valves has the further great incidental advantage that the application of pressure to the power piston of the main valve may be closely regulated by choke and metering valves,

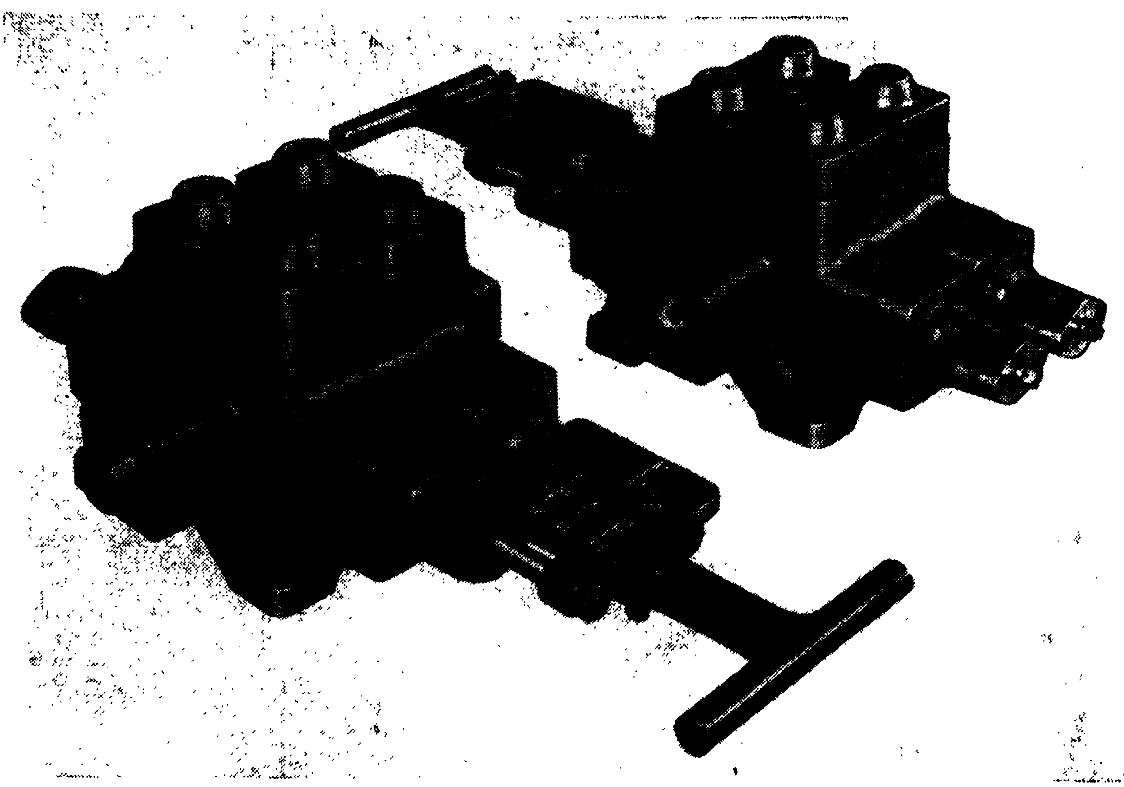


FIG. 176. Change-over valve. (*The Oilgear Co., Milwaukee, Wis.*)

so that smooth and closely controlled operation of the main valve may be had, regardless of the speed with which the pilot valve is operated. Pilot valves are miniature editions of directional-control valves and may be made as balanced-piston valves of the two-, three-, or four-way type or as rotary plug valves. In addition to the operation of master valves, they have found a multiplicity of uses. Figure 153 shows the application of a pilot valve as vent valve. Pilot valves lend themselves well for automatic operation of master valves by operating the pilot valves with dogs from the moving parts of the machine. A separate source of pilot pressure may be used, or the valves may derive their supply of operating fluid from the main source of supply. In that case, installation of a low-pressure sequence valve may be necessary, with the pilot valve between

the source of pressure and the sequence valve to ensure availability of pilot pressure at all times.

One of the most universally applicable types of pilot valve is the rotary valve manufactured by Vickers Inc. This versatile unit, made in ¼-in.

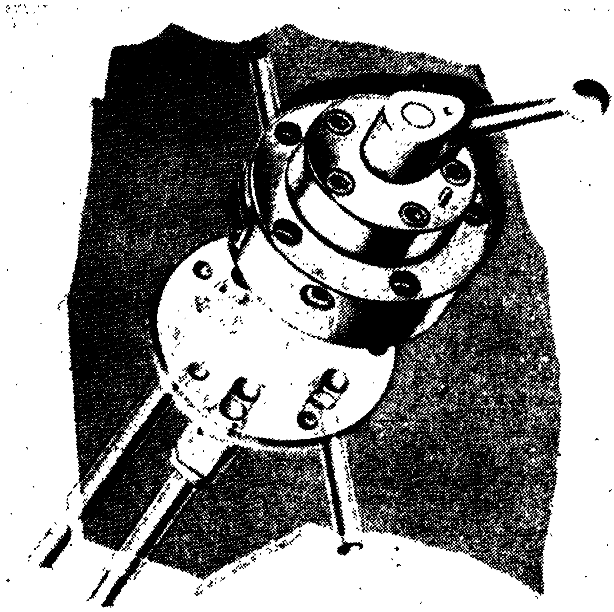


FIG. 177. Rotary pilot valve. (*Vickers Inc., Detroit, Mich.*)

pipe size for a maximum flow of 3 gpm at 1,000 psi, is shown in Fig. 177, in a flange-mounted design. The valve may be supplied with a number of mountings, including tapped mounting holes, flange mounting, surface mounting, and gasket mounting. In the gasket-mounted valve, the four outlet connections are brought out through the back surface of the valve

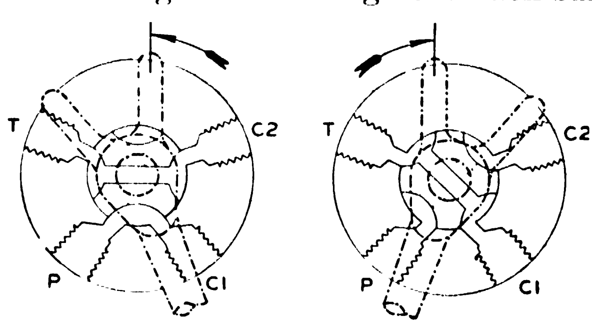


FIG. 178. Diagrammatic section of Vickers rotary pilot valve.

and individually supplied with Vickerseals. Pipe connections are made in the back mounting plate, and the valve may be removed without disturbing the piping. A number of Vickers units are now being made in this manner. Several styles of hand levers are available, such as reversing dogs, ball handles, and star wheels.

Operation. Figure 178 shows a diagrammatic section of the Vickers pilot valve. Revolving the stem of these valves rotates the spool to

direct the flow of oil suitably. The revolving spool contains two passages. When the valve is used as a pilot reversing valve with lever at counterclockwise position, oil from the pilot pressure source will flow through the spool to (as an example) one end of a pilot-operated four-way valve, while oil from the opposite end of the four-way valve will return through the spool to the tank. Revolving the spool to clockwise position will direct the flow of the pilot circuit to the opposite end of the four-way valve, while oil from the end to which it flowed in counterclockwise position will return through the spool to tank. This action causes the

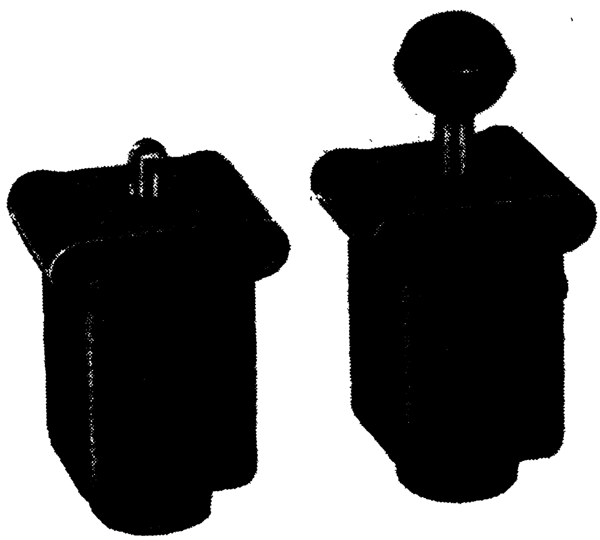


FIG. 179. Roller-actuated pilot valve. (*Vickers Inc., Detroit, Mich.*)

four-way valve to reverse the flow of the main hydraulic circuit, whenever the pilot-valve spool position is changed.

When the valve is used as a two-way valve or for other special functions, one or two connections may be plugged. For instance, when venting Vickers Hydrocone relief valve, either connection 1 or connection 2 serves as the pressure inlet, while the "tank" connection is piped to the oil reservoir and the other two connections are plugged. Valves may be furnished with external drains when the tank connection is subjected to pressure at any time. This is the case when the star-wheel lever is used. With this device the valve may be indexed continuously in one direction, and each one-eighth turn reverses the pilot circuit. The operating arc of the valve is 45°. The neutral or crossover position is at $22\frac{1}{2}°$. All ports are blocked in that position.

Two valves may be mounted in tandem, operated by a single lever for operating combined and interlocking functions. The valve may be supplied with double solenoids for electrical actuation. The small compact unit is about 3 in. in diameter and $3\frac{3}{4}$ in. long and weighs 4 lb.

Piston-type Pilot Valves. Pilot valves are available in balanced-piston design, very similar in construction to the directional-control piston, valve previously discussed. They are made in two-, three- and four-way types and may be cam or roller operated, manually or electrically

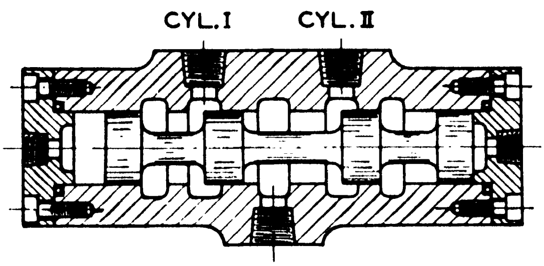


Fig. 180. Pilot-operated valve.

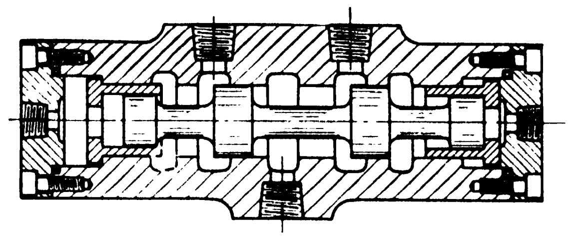


Fig. 181. Self-centering pilot-operated valve.

actuated. Generally ¼-in. pipe connections are used for this type of valve. Figure 179 shows the Vickers piston-type pilot valve, roller-actuated.

Operating valves may be pilot actuated by provision of actuating pistons in the end housings. This may be done in the manner shown in Fig. 180, where a simple single-acting piston is employed at each end of the valve plunger. Applying pilot pressure to either end and exhausting the opposite end will cause the main valve to move "hard over" in the corresponding direction. It is well to remember that with this type of operation, only two positions of the main valve are obtainable, hard over in either direction. By employing a centering spring and using a pilot valve with both cylinder connections exhausted in neutral (see Fig. 168, second from bottom), a third or mid-position may be accomplished. The same result may be obtained by the use of self-centering pistons (Fig. 181) in combination with a center-pressure pilot valve (Fig. 168, bottom row).

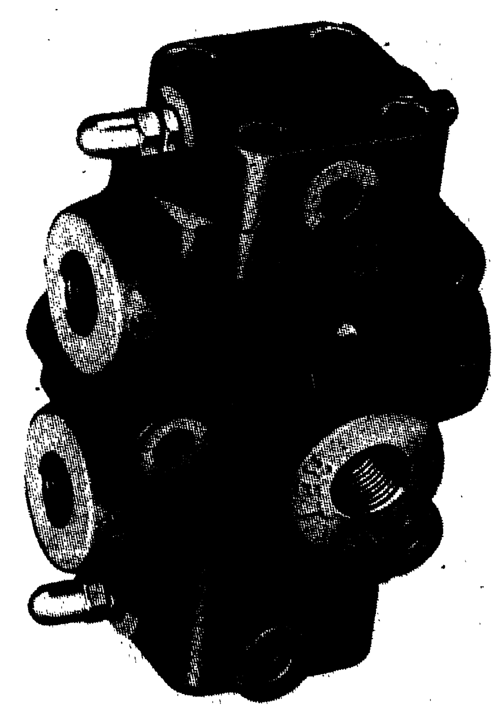


Fig. 182. Pilot-operated valve. (*Vickers Inc., Detroit, Mich.*)

Intermediate or throttle position cannot be obtained with pilot-operated valves, but close control of valve operating speed may be had by use of choke checks, as in Fig. 156, installed at both ends, whereby the

speed of the valve plunger in either direction may be independently adjusted. Sometimes these choke devices are built in the valve end housings. Adjustable stops may be provided, if desired, to adjust the end location of the valve plungers. Figure 182 shows a Vickers operating valve with pilot chokes built in the end housings.

Solenoid-operated Valves. Control of hydraulic power by electric push buttons has become more and more popular on modern machine tools. Electrically controlled hydraulic power requires the use of electrically actuated valves. For this purpose, solenoid-actuated valves have been developed. These valves are available with all possible combinations of plunger functions, as illustrated in Fig. 168, and may be supplied in single-solenoid, spring-opposed models and double-solenoid models, both in spring-centered designs and without springs. The single-solenoid type will result in a two-position valve: hard over in one position with solenoid deenergized and hard over in the opposite direction with solenoid energized. The double-solenoid type may be three-position valves with a spring-centered neutral position, and two end positions obtained by energizing one or the other of the solenoids. A number of problems peculiar to the operation of hydraulic valves were encountered and had to be solved before foolproof and trouble-free solenoid-actuated valves could be placed on the market. In the following, a few hints for the design of these valves will be given, together with a sample calculation for one of these valves, which will aid the designer in avoiding some of the pitfalls of this seemingly so simple design problem.

We have mentioned before that despite greatest care in manufacture and excellent finish these valves exhibit a tendency to stick or lock, which increases with increasing pressures and tends to become aggravated when the valve is held under pressure for extended periods of time. The valve must be so designed that it will readily break loose when the electric power is disconnected. This involves proper selection of springs of sufficient strength to do this under all operating conditions. The solenoids must have sufficient strength to operate the valves against the spring tension and also to break the plungers loose in locked position. The following practical hints may be found useful. To minimize locking effect, good surface finish on pistons and bores is essential, as well as proper grooving of pistons. Accurately machined grooves of U section and constant width are necessary. Packings at the ends of the valves must have a minimum of friction. O rings installed according to recommendations of Chap. VIII are very satisfactory.

Direct solenoid-actuated valves are satisfactory up to about 1-in. size. This will indicate pistons of about $1\frac{1}{4}$-in. diameter and $\frac{3}{4}$-in. valve

stroke. Larger valves should be pilot operated. Maximum spring pressures should be from about 15 lb for a ¼-in. valve to about 30 lb for a 1-in. valve. Minimum spring pressures to move valve at zero pressure are from 7½ to 15 lb. Solenoids are made by a number of reliable concerns and should be carefully chosen for their electrical and mechanical characteristics. Solenoids should be capable of operating the valves at 15 per cent below and above their rated voltage. Alternating-current solenoids have almost entirely superseded d-c solenoids, which should be strictly avoided for hydraulic control valves. The pull of a solenoid is proportional to the square of the voltage, and this fact must be given due consideration when specifying the pull required to operate the valve at

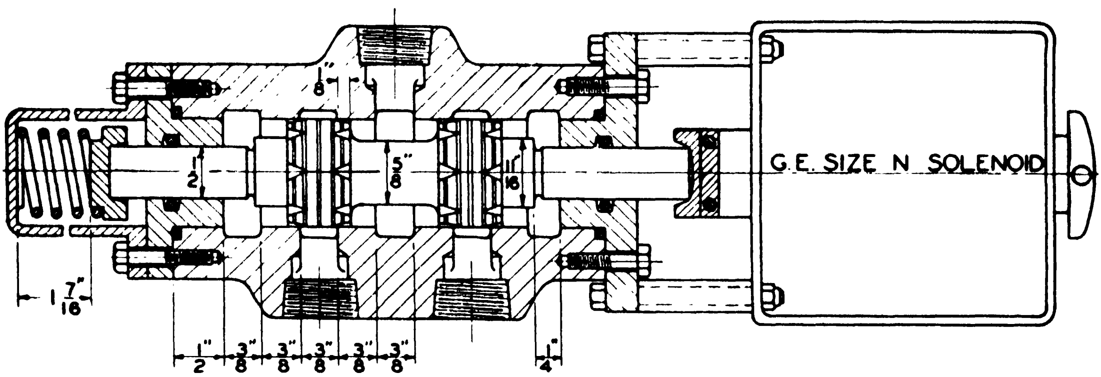


FIG. 183. Design of solenoid-operated ¾-in. four-way valve.

reduced voltage. Great care must be taken to make sure that the solenoid will not overheat when left in the "on" position for long periods, as required in the duty cycle of many hydraulic machines. Recommendations from the manufacturer of the solenoids should be secured. Temperature rises in general should not exceed 85°C. A similar condition exists in rapid-reciprocating duty cycles, where the integrated power input should not exceed that specified for safe operation by the manufacturer. Strokes available on standard solenoids generally exceed those required to operate the valves, and the expedient would seem obvious to use some sort of linkage to gain advantage in pull and get by with a smaller solenoid. It will be found that, in general, it is better practice to use direct-acting push-type solenoids of large enough size and utilize only part of their available stroke. Any linkage or lever mechanism will soon batter itself to pieces under the severe operating conditions to which these valves are subjected. For the same reason, pull-type solenoids should be avoided.

Example: To acquaint the reader further with some of the problems encountered in the design of solenoid valves, a calculation will be given for a ¾-in., four-way, spring-opposed solenoid valve. The valve is to have a 1-in.-diameter piston, ⅛-in. lap, and ⅛-in. port opening, which results in the design shown in Fig. 183.

We lay the valve out with ⅜-in. port slot width and ⅜-in. land space between port slots. The plunger is shown in neutral or mid-position. Lap each side of port slot is ⅛-in. Valve is moved ¼-in. each side of neutral, resulting in ⅛-in. port opening. The valve piston is shown with center by-pass having a total of 16 V notches, 45° apex angle in each piston, extending ⅛-in. into the port slot. In hard-over position, the V grooves are lapped ⅛-in. The area of the valve is 0.392 sq in., and the combined by-pass area of V slots is 0.10 sq in., giving a velocity ratio of 4:1. With 10 ft per sec speed through the valve, the speed through the V slots in neutral would be 40 ft per sec. We select a spring giving 25 lb maximum pressure with valve in extreme position and 10 lb minimum pressure with valve in opposite extreme. This results in a spring gradient of 30 lb per in. A spring of ¾-in. pitch diameter and No. 13 W. & M. wire gauge will be found satisfactory, having a Wahl index of 8:1.

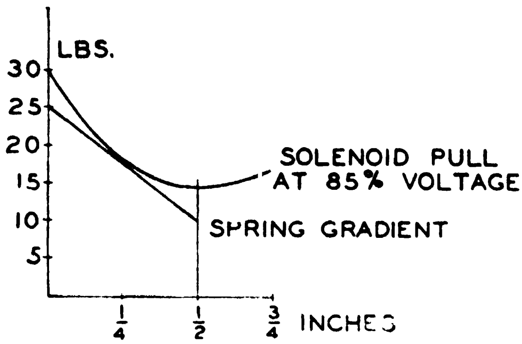


FIG. 184. Pull curve of General Electric size N solenoid and spring gradient.

Maximum stress (uncorrected) is 60,000 psi, and eight acting coils are required. Initial compression is approximately 5/16 in. and free length 2 in. For operation of the valve, we select a General Electric CR 9503, size N, push-type solenoid.

Figure 184 shows the pull curve of this solenoid at 85 per cent voltage and the spring-pressure gradient. In determining the solenoid, the fact is of interest that in hydraulic-valve operation, the force exerted by a solenoid seems much more effective than an equal force exerted by other means, such as a spring or weight. This is probably due to the 60-cycle force oscillation, which results in a great number of high peaks and low valleys, the integrated average of which is the pull rating. This rapid oscillation undoubtedly reduces the frictional resistance, but is, of course, ineffective against a steady load such as the valve spring. Therefore, if the solenoid rated force is comfortably above the valve-spring gradient, it will always move the valve plunger.

TABLE III

Time on, per cent	*Operations, per min*
25	74
50	55
75	36

Moreover, in valves without spring return, much smaller solenoids may be used and will almost always break a pressure-locked valve loose. The General Electric solenoid is furnished with a pin connection between the push bars and arranged for 1½-in. stroke. Since only ½-in. stroke is used for this application and no mechanical connection between valve plunger and solenoid is made, the bar extension is modified, as shown, and a T-shaped steel piece is inserted and riveted between the bars. The solenoid is solidly mounted upon the valve end cap with four 5/16 cap screws and spacing sleeves. Lock nuts of standard design complete the assembly.

The solenoid will withstand full voltage indefinitely when seated. The operating cycles shown in TABLE III are permitted with this solenoid. This is satisfactory for almost any hydraulic application.

Solenoid-pilot-operated Valves. Large valves that would require ungainly solenoids for operation will best be pilot operated. To this end, regular pilot-operated valves may be operated remotely with solenoid pilot valves. A number of combinations is possible for operating these valves. Simple pilot plunger valves, as shown in Fig. 180, may be operated by solenoid-actuated pilot four-way valves, which results in two

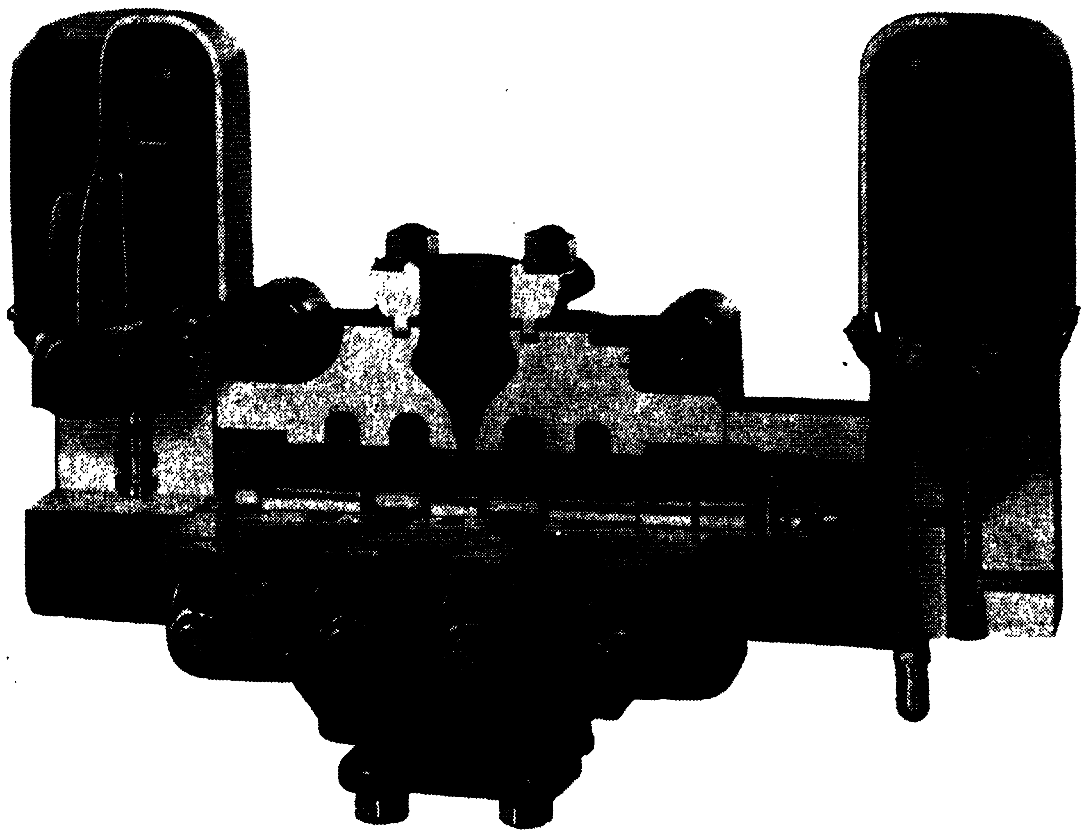


FIG. 185. Solenoid pilot-operated four-way valve. (*Vickers Inc., Detroit, Mich.*)

positions of the pilot-operated valve. The same valve may be made with a centering spring and actuated by a spring-centered, double-solenoid pilot valve, or by two spring-opposed, three-way, solenoid pilot valves, resulting in three positions. Self-centering valves, as shown in Fig. 181, may be operated by center-pressure, double-solenoid, spring-centered pilot valves, or by two solenoid three-way pilot valves. Main and pilot valves may be built in one unit, as shown in Fig. 185, which illustrates a Vickers spring-centered, pilot-operated, four-way valve, with built-in three-way solenoid pilot valves and adjustable choke checks.

Solenoid-operated valves, both for direct and pilot operation, are made by all manufacturers of hydraulic components. Vickers makes single-

and double-solenoid-operated valves, direct solenoid operation being used on the ½- and ¾-in. valves, and pilot valves being used on the larger valves, as shown in Fig. 185. These valves are rated at 1,000 psi.

Hydro-Power Inc. makes a complete line of solenoid-actuated valves in sizes from ½ to 1¼ in., for 2,500 psi pressure. Valves may be supplied in single-solenoid spring-opposed and double-solenoid spring-centered types.

Racine Tool and Machine Co. make their valves in pilot-operated models in all listed sizes. Gerotor May Co. valves are available in solenoid-operated models up to 1 in. in size.

Two- and Three-way Valves. Obviously the simple balanced-piston operating valve lends itself well to operation as a two-way or shutoff or

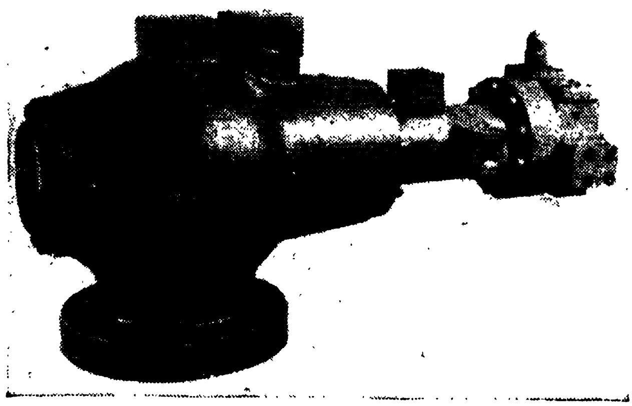


FIG. 186. Prefill valve. (*The Oilgear Co., Milwaukee, Wis.*)

three-way valve by merely leaving off or plugging pipe connections. When this type of valve is used as two-way valve, a drain connection must be provided to drain off leakage past the pistons. These valves, as well as the four-way valves, may be mechanically actuated by cams and rollers. In that case, they may serve as decelerating valves. Decelerating valves are sometimes made with built-in checks to permit full return flow. A special form of two-way valve is the prefill valve, often used on presses and other large machines. This valve, mounted between a surge or prefill tank and the hydraulic cylinder it serves, permits filling the cylinder with oil, while the piston is advanced rapidly by means of a small so-called "booster" or "kicker" ram or by gravity. On encountering resistance, the ram slows down, and the prefill valve closes by spring pressure or an auxiliary hydraulic plunger, permitting the pump to build up pressure in the cylinder. On the return stroke, the valve is forced open by another hydraulic plunger, permitting escape of oil from cylinder, while rapid return of the ram takes place by the pump delivering into the rapid-return cylinder. Velocities in these valves may be made from 10 to

15 ft per sec, provided that there are no extended pipes between valve and tank and/or cylinder.

Figure 186 shows the Oilgear prefill valve. This valve has a built-in resistance or sequence valve, used when booster rams are employed. Upon building up of a fixed predetermined pressure on the booster rams, this valve opens and permits shifting the prefill valve to closed position, at the same time the flow of oil from the pump is directed into the main

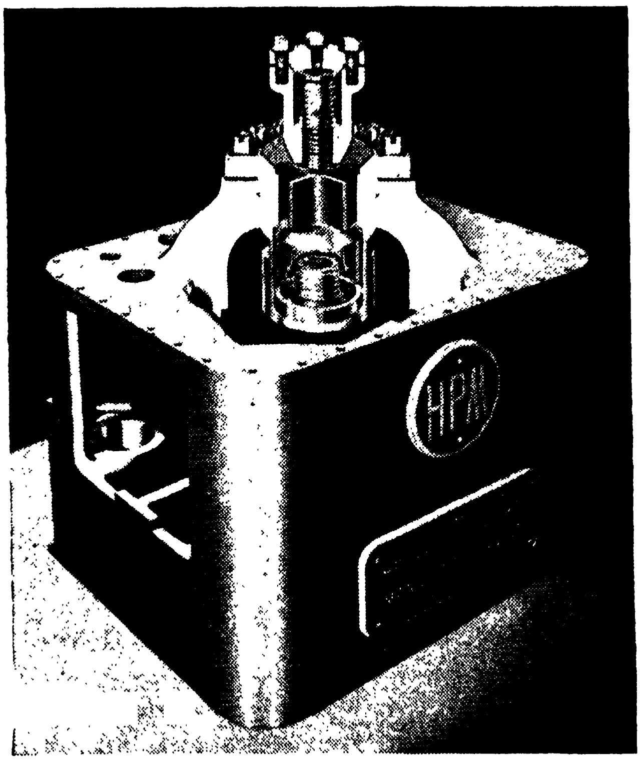


FIG. 187. The dome of a hydraulic-press cylinder, cut away to show the surge valve. (*The Hydraulic Press Mfg. Co., Mount Gilead, Ohio.*)

cylinder. The valve may also be used on gravity-operated presses. Often these valves are made as overhead check valves, directly connecting the cylinder with a tank mounted above it.

Figure 187 shows the arrangement used by the Hydraulic Press Mfg. Co., Mount Gilead, Ohio. In this case the ram of the press advances by gravity, while the oil flows into the cylinder through the opening provided by the spring-suspended check-type valve. As soon as resistance is encountered by the ram, the check closes owing to the action of the supporting spring, and the pump, which is permanently connected to the cylinder, builds up pressure. Upon reversal of the press, pressure is supplied to the operating plunger mounted above to force open the surge check and permit escape of oil from the press cylinder.

Check or nonreturn valves are frequently used in hydraulic systems.

The simplest valve of this type is made by employing a hardened steel ball in a cast or forged housing. This type of valve is very satisfactory, but not always quiet in operation. Smoother and quieter operation may be had with spring-loaded plunger-type check valves, as made by Vickers Inc. and others. Vickers check valves are available in all pipe sizes up to 2 in. They are frequently used built into other valve devices to form various combinations of piston and check valves.

4. VOLUME CONTROLS

We have seen how the magnitude of hydraulic pressure is governed by pressure controls, and how hydraulic pressure is directed to the point of its application by directional controls. In the following we will deal with the devices that control the quantity of flow to a circuit or to parts of the circuit.

Quantity control of fluid flow is of great importance in most hydraulic applications. We have seen in Chap. VII how the output of a variable-delivery pump is regulated by manually adjustable or automatic controls. The advantage of this type of pump has been discussed; it lies in the possibility of automatically adapting its delivery characteristics to predetermined requirements and in the low power consumption at partial loads, due to the adjustable stroke characteristics. Variable-delivery pumps are high-priced pieces of machinery and for this reason cannot be used on many applications, where the employment of low-priced, low-pressure, constant-delivery pump is indicated. Recognition of this situation has led to the development of numerous speed or volume controls, functioning with great accuracy, which supply the same degree of close speed regulation as the variable-delivery controls and under certain conditions of operation approach them in efficiency. These controls, which go under the general name of "throttle controls," will be described in the following.

The simplest way of controlling the volume supplied by a constant-delivery pump, would obviously be the introduction of a simple throttle or needle valve either to bleed off a certain desired volume or to restrict inflow or outflow of the hydraulically actuated cylinder. This means of speed control has the serious disadvantage that with variation in the work load the pressure drop through the throttle is not constant, and the flow through the throttle, being a function of the pressure drop, will, therefore, vary as a function of the work load. This form of volume control, therefore, is not suitable for applications where there is a variation in work load, and has largely been abandoned for all but a few applications, where the load is fairly constant and moderate variations in speed are not objectionable. The disadvantage of the simple throttle valve may be overcome by the provision of a fixed pressure drop across the adjustable

throttle, regardless of work resistance, which produces a reasonably constant flow through the throttle at a given setting. This may be accomplished easily by introduction of a constant-reduced-pressure reducing valve (Fig. 163). Regardless of pressure variation at the intake of the

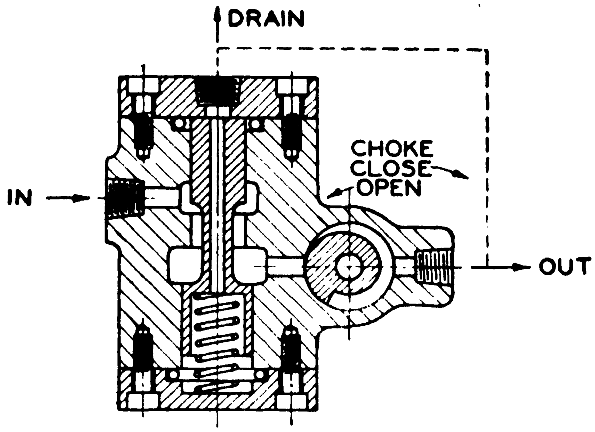


FIG. 188. Metering valve with constant pressure drop.

reducing valve, the output pressure will always be constant and will produce a constant flow through an adjustable throttle. Figure 188 shows how this may be done. In the form shown in this illustration, the valve may be used as bleed-off or metering-out valve. To operate as metering-in valve, all that is necessary is to connect the drain opening

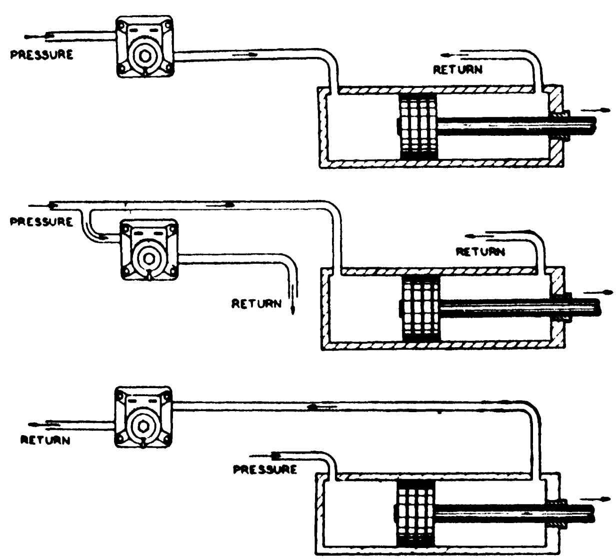


FIG. 189. Three hookups for metering valves. (*Vickers Inc., Detroit, Mich.*)

in Fig. 188 to the outlet of the throttle valve, as shown in dotted lines. The bleed-off pin in Fig. 163 has been omitted in Fig. 188, as it is not required for this application. In commercially available valves, the connection between outlets, shown in Fig. 188, is built into the unit, so that the valve may be hooked up in any one of the three combinations. Figure 189, supplied by Vickers Inc., shows these hookups and recom-

mendations for their use. The reduced pressure or pressure drop across the choke is generally made quite low, in the neighborhood of 40 to 50 psi. This results in large throttle openings, free from clogging by small particles. The valve becomes ineffective when the intake pressure drops below the throttle pressure.

In a metering-in valve of the type described, the pressure of the pump is always at the maximum, determined by the relief valve setting, regardless of work resistance. The efficiency of this kind of arrangement is extremely low, as all of the pump output not used for operating the motor passes through the relief valve at maximum pressure, and even that part which does useful work is subjected to throttle losses, until the work

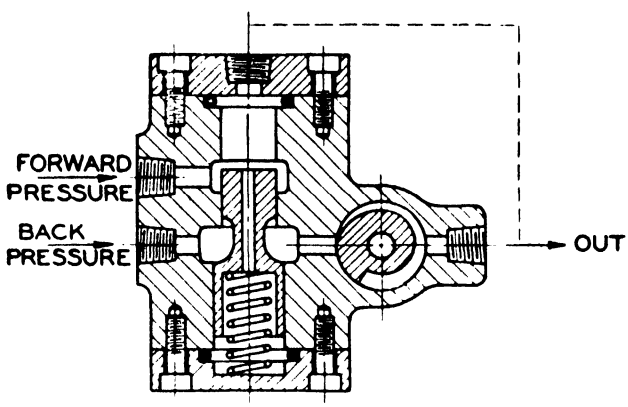


FIG. 190. Back-pressure valve with constant back pressure.

resistance approaches the relief-valve setting. Therefore, the use of this arrangement should be avoided with constant-delivery pumps, but is well adapted for applications of accumulators or variable-delivery pumps with pressure-compensating controls. Losses in both these cases are confined to the throttling loss through the metering valve. Compared with a simple variable volume control on a variable-delivery pump, the arrangement has the advantage of automatic slip compensation. Bleed-off valves are more efficient, inasmuch as the part of the output of the pump that is not used is by-passed at the actual working pressure existing in the system instead of at the peak pressure, and there is no throttling loss on the fluid doing work. Obviously this type of arrangement cannot be used where the working pressure is below the throttling pressure of the control. In some types of light machine-tool drives this may be the case.

When the valve is used as meter-out or back-pressure valve, a back-pressure circuit will result, which is desired for many machine-tool applications such as drilling, milling, etc. In this application the pump pressure is again constant, regardless of work resistance, while the back pressure drops with increasing work resistance. The efficiency of this

circuit again is very low, but a negative work load of practically any magnitude may be supported.

Better efficiency with a back-pressure circuit may be obtained by holding the back pressure constant and letting the feed pressure vary with the work load. This may be done by a modification of the valve structure, as shown in Fig. 190. The disadvantage of this arrangement is that it cannot support negative work loads in excess of the throttle pressure.

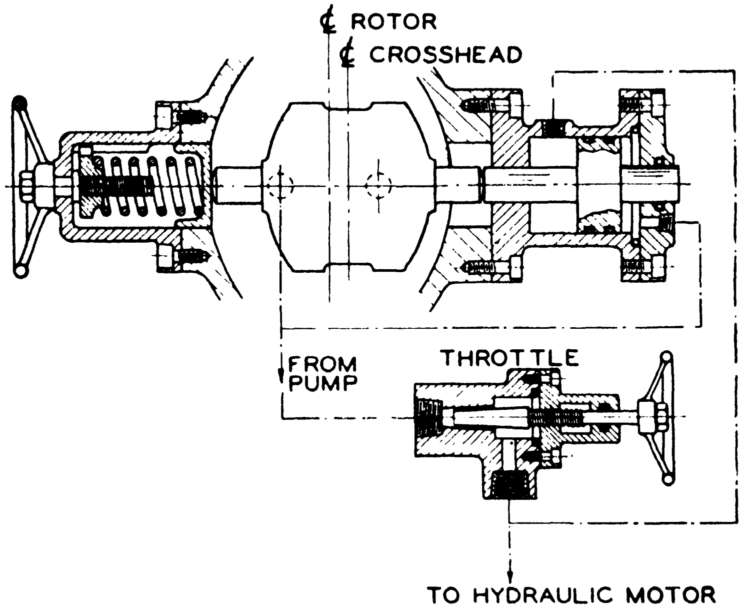


FIG. 191. Metering-in control for variable-delivery pump.

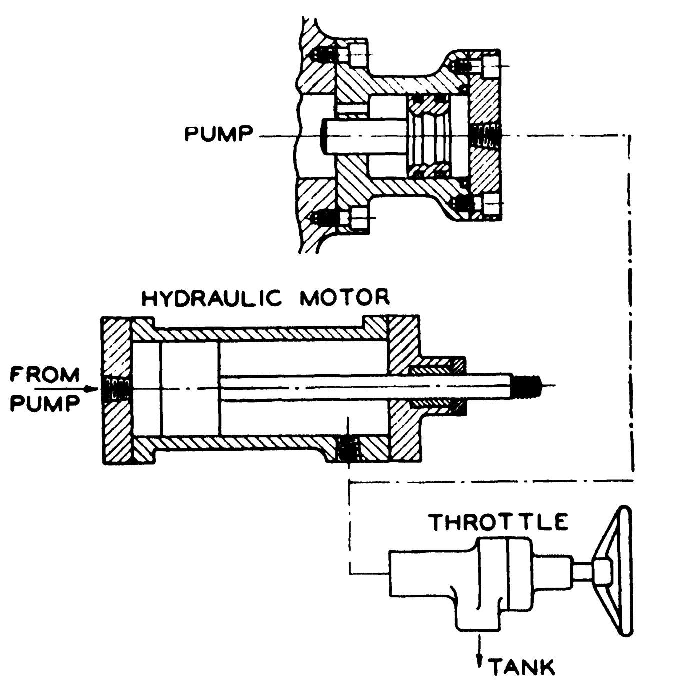


FIG. 192. Metering-out control for variable-delivery pump.

It is of interest to note that both arrangements (Figs. 188 and 190) may have their counterparts in a variable pump control, whereby the pump control takes the place of the reducing-valve unit and the pump output is controlled by adjustable chokes. This arrangement is illustrated in Figs. 191 and 192, showing forward or meter-in control and back-pressure control. The latter will not support negative work loads in excess of throttle setting. Both automatically compensate for slippage variations in the pump.

FIG. 193. Metering or flow-control valve. (*Vickers Inc., Detroit, Mich.*)

Commercially available metering valves are made by Vickers Inc. in a large range of sizes and styles. Operating principle is the same as that shown in Fig. 188, and the valve may be hooked up in any of the combinations shown in Fig. 189. Valves are available in panel- and gasket-mounted styles and in models with built-in relief valves. The neat and compact design of the device is illustrated in Fig. 193, showing panel-mounted units. The choke or throttle is of the eccentric-slot type, illustrated in Fig. 188, and a full adjustment scale covers about 90° turn of the dial.

Flow Dividers. In the preceding paragraphs we have shown how a constant flow of oil, regardless of resistance, may be produced by establishing a fixed pressure drop across a choke. The fixed pressure drop was produced by means of a spring-loaded valve, whereby the setting of the spring determined the magnitude of the pressure drop. It is possible to eliminate the spring and balance two pressures against each other. This principle is used in the Bendix flow equalizer, shown in Fig. 194. In this flow equalizer the fluid passes through the inlet port into the orifice sleeve. Here it meets the obstruction of a plug that moves downward

when sufficient pressure is built up. This uncovers two opposite holes in the sleeve, through which the fluid splits, and flows down the side passages *A* and *B*. The fluid forces the check valves open, and the metering piston then passes the fluid out through two metering grooves in the piston circumference. The returning fluid reverses the above operation, except that the fluid enters a second pair of metering grooves in the piston and reaches the two side passages through a second pair of check valves. The fluid then enters a small slanting passage to the bottom of the orifice sleeve, forcing the sleeve upward, opening the two orifice holes. The automatic equalizing of the two streams depends on the fact that any difference in the rate of flow between the side passages *A* and *B* causes a corresponding difference in pressure loss. The metering piston moves to the low-pressure side to equalize this pressure difference. This action throttles the flow of fluid through that side of the metering piston until equal flows have been obtained.

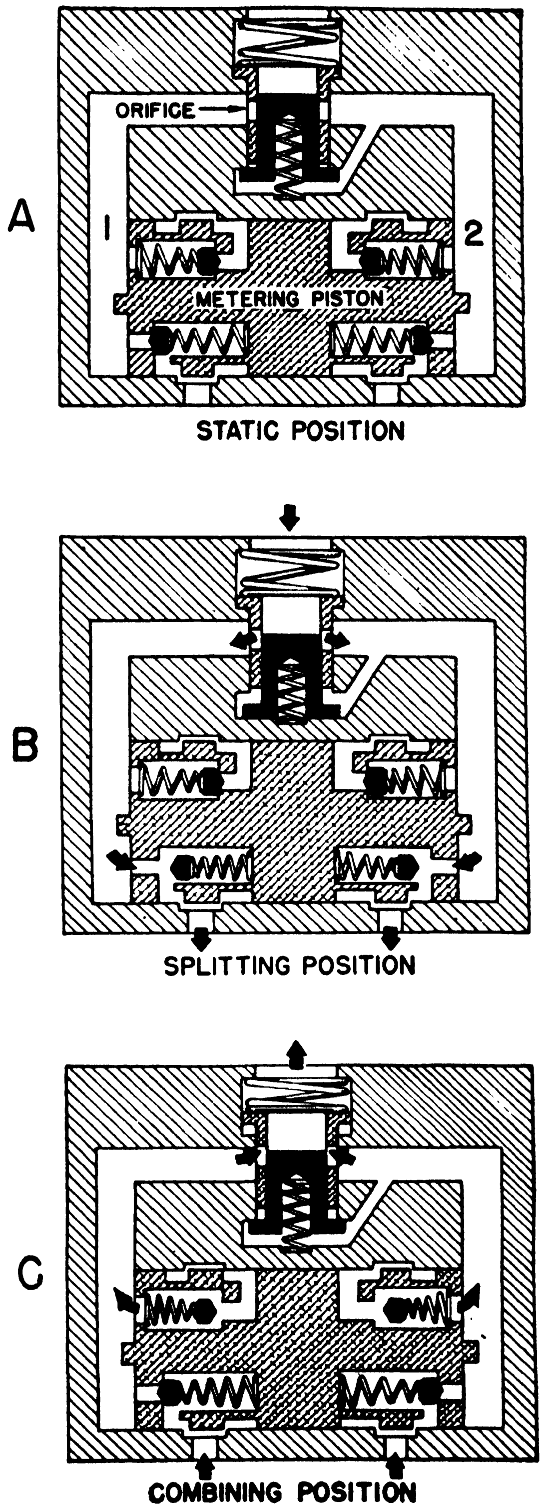


Fig. 194. Bendix flow divider. (*Bendix Aviation Corp., Pacific Division, North Hollywood, Cal.*)

5. COMBINATION CONTROLS AND PANELS

We have explained in the beginning of this chapter that different control means may be combined in one unit to produce a combination of stipulated functions. For instance, pressure- and directional-control units may be built together to control a hydraulically actuated machine. Likewise, combinations of volume and directional controls, volume and

pressure controls, and other more involved hookups may be made, so that it will be possible to group all the functional units of a hydraulic system in one central control panel. Standardized control panels are offered by a number of manufacturers to permit automatic operation of certain machine tools and other hydraulic devices, and a few representative units will be described in the following.

Vickers Inc. This company offers a number of standardized machine-tool panels for manual, semiautomatic, and automatic control of tool

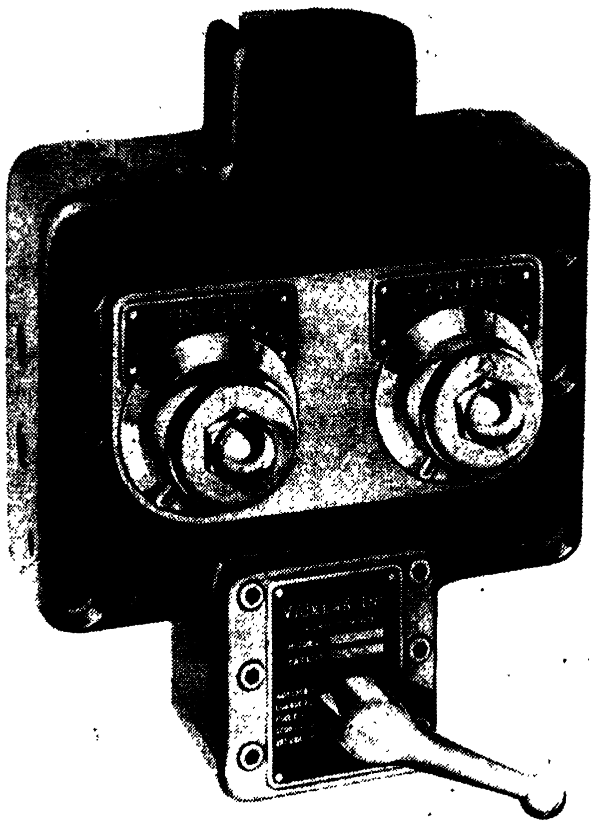


Fig. 195. Vickers feed-cycle control panel. (*Vickers Inc., Detroit, Mich.*)

slides. The feed-cycle control panel is shown in Fig. 195. Essentially this unit consists of a multiple-piston valve that may be actuated by the lever shown in front of the assembly and has a roller extension at the top, concealed by the apron shown at the top in Fig. 195. This roller extension may be actuated by cams, mounted on the machine-tool slide, so as to occupy the following positions, reading from the top down:

Rapid advance
Coarse feed
Fine feed
Stop
Rapid return

Coarse and fine feed are produced by two metering-out flow-control valves, the dials of which are visible in front of Fig. 195. The rapid-advance stroke is initiated by operation of the hand lever. The table proceeds at high speed produced by the full pump capacity, until a dog mounted on it depresses the control-valve plunger, producing the coarse feed. The table continues at the coarse-feed rate, until the fine-feed cam is engaged, producing the fine-feed rate. Finally the table reaches the end of the feed stroke, where a limit switch is engaged, energizing a solenoid that produces the return position of the valve plunger. It must be kept in mind, when applying these panels, that the return position of the valve plunger must be initiated by a force or movement other than that derived from the machine head or element being propelled by the hydraulic cylinder. To this end, Vickers supplies models with lever shafts extended to the rear for attachment of solenoid-actuated levers, or units with built-in solenoids or pilot cylinders may be furnished, providing for the automatic-reverse function. After completion of the rapid-return stroke, a hook-type cam is utilized to return the plunger to the "stop" position. In addition to models equipped with a single return solenoid, double-solenoids units are available for remote-control initiation of the rapid forward stroke, as well as pilot-actuated models. Application of these panels to control circuits will be illustrated in Chap. XI.

Two-directional feed panels are available with rapid traverse and feed in both directions. In addition to this, the company supplies a completely electrically actuated feed panel, controlled entirely by limit switches, providing automatic or manual start, rapid advance, feed, rapid return, and stop. Automatic cycling panels with pilot valves, operated by table dogs, may also be supplied. The number of possible combinations is practically unlimited, and special panels having any number and combination of units may be built up. Panels are designed for gasket mounting, all pipe connections coming out of the back of the unit, and are provided with Vickerseals for attachment to the machine surface.

Sundstrand Machine Tool Co. The PWX Sundstrand Feed Unit. *Description.* The Sundstrand PWX feed units (Fig. 196) consist of a constant-displacement and a variable-displacement multiple-piston-type pump, feed-adjusting mechanism, and the main control valves, all in a single housing.

Variable-displacement Pump. The feed pump is a variable-displacement pump of the multiple-piston type. There are five pistons, each with an intake and outlet check valve. With the piston chamber filled, the piston forces the oil out through the outlet check valve to feed the main cylinder. The inlet valve automatically closes, so that all trapped oil is used for feeding the main cylinder. The pump provides one or two

rates of feed, both adjustable from zero to maximum. (Some pumps are especially arranged for three feed rates.) The pump was described in detail in Chap. VII.

Feed Adjustment. The two feed rates are adjusted by the two knobs on the outside of the pump. One is for fast feed, the other for slow feed. The two knobs turn worms that rotate cams, one cam providing the setting for fast feed and the other for slow feed. A wobbler-support

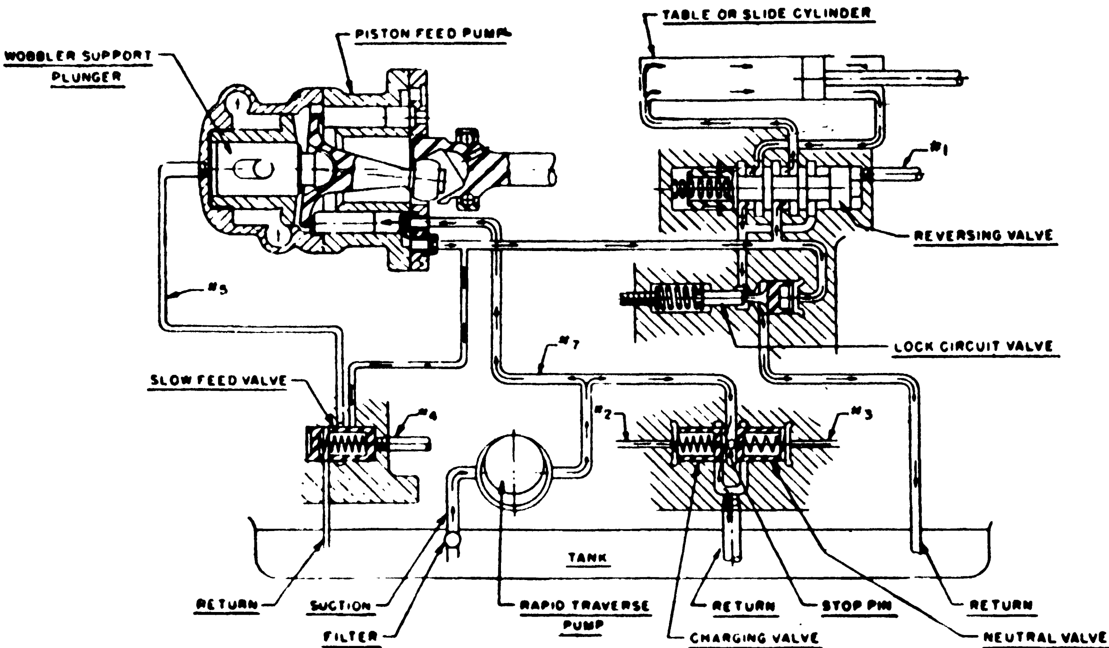


FIG. 196. The Sundstrand Machine Tool feed circuit. (*The Sundstrand Machine Tool Co., Rockford, Ill.*)

plunger carries a pin that registers against either one cam or the other to give either the fast- or slow-feed rate.

Constant-displacement Pump. Rapid-traverse actions are produced by the constant-displacement pump. It also furnishes the oil for charging the variable-displacement-piston pump. The constant-displacement pump is a self-priming rotary pump consisting of an external rotor, internal roller, and crescent.

Main Control Valves. The main control valves are in the PWX pump housing and are, in turn, controlled by pilot or solenoid valves. Following are the main valves:

BACK-PRESSURE VALVE. The back-pressure valve is on the return side of the main cylinder. It produces a slight back pressure, enough to provide a steady feed under no load. This valve also blocks the return of oil to the tank in a climb cut or when a drill breaks through the work. It is a simple valve that is opened by the working pressure and closed by a spring. The spring determines the minimum (but not maximum)

working pressure, so there is always a certain fixed minimum working pressure. The spring also controls the amount of opening required in the valve during a climb cut to keep the piston in the main cylinder from traveling faster than the rate the piston pump is set for. During a climb cut, as on a milling machine, there is no need for the piston pump to build up any pressure, because the milling cutter pulls the work, forcing the oil out of the main cylinder.

Reversing or Four-way Valve. The four-way valve is between the main cylinder and the locked-circuit valve. It is positioned at one end of the valve bore by means of a spring, creating forward cylinder movement by directing oil to one of the outlet ports. Reverse movement occurs when hydraulic pressure is exerted against the plunger end opposite the spring end, the oil being directed in the other outlet port.

Neutral Valve. The entire output of the rapid-traverse pump returns to the tank when the neutral valve is open. When in feed and rapid traverse, the valve is closed and remains closed during the entire cycle. There is a small hole in the valve plunger that permits oil to return to the tank through a pilot line to the pilot valve. Closing off the escape of oil through this hole with the pilot valve closes the neutral valve.

Charging Valve. When open, this valve creates the pressure necessary to charge the variable-displacement-piston pump with oil from the rapid-traverse pump and allows the surplus oil to return to the tank at the charging pressure. When closed, the entire output of the rapid-traverse pump is forced through the check valves in the piston pump to the actuated unit, producing the rapid-traverse rate. There is a small hole in the valve plunger that permits oil to return to the tank through a pilot line to the pilot valve. Closing off the escape of oil through this hole with the pilot valve closes the charging valve.

Slow-feed Valve. This valve determines which of the two set rates of feed will be obtained. When it is open, fast feed is produced, and when it is closed, slow feed is produced. Where a third rate is used, this valve must be open during the intermediate feed.

Relief Valve. A relief valve is connected to the pump-pressure-supply source and acts as a safety valve during overload or when the actuated machine member feeds against a positive stop. It is normally set for a pressure of 1,000 psi.

Pilot Valves. The main valves in the PWX pump housing require pilot valves such as the 14X and 16X Sundstrand valves or Sundstrand electrical-solenoid valves to govern them. These auxiliary valves are connected to the main control valves by pilot lines and are actuated by dogs on a moving member of the machine. The dogs trip the pilot valves or open and close limit switches to operate the electric solenoid valves.

The 16X pilot valve is shown in Fig. 197; it provides feed in two directions. The 14X valve provides feed in one direction only.

Lines 1 and 2*A*, one connected to each end of the four-way valve, control the position of the valve stem. Actuating the pilot or solenoid valve will admit pressure to one of the lines and simultaneously open the other to the tank. This forces the four-way-valve stem to the position desired. The 2*A* line is found only on the 6PWX and 11PWX pumps. In the 5PWX and 10PWX pumps, the four-way stem is held in position for forward travel by a spring, and only line 1 connected to the four-way

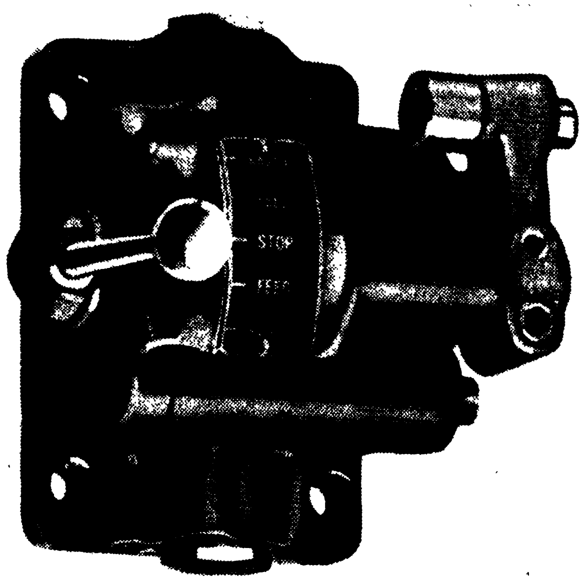


FIG. 197. The 16X pilot valve. (*The Sundstrand Machine Tool Co., Rockford, Ill.*)

valve is necessary. Pressure in line 1 overcomes the spring resistance and shifts the valve stem to position for rapid return. When line 1 is opened to the tank, the spring forces the valve stem to its former position.

Line 2 connects the spring end of the charging-valve chamber with the pilot valve. Closing line 2 with the pilot valve causes the valve plunger in the charging valve to seat. This is done by preventing the oil from escaping through the line 2 to the pilot valve and back to the tank. When line 2 is open, the charging valve is open, and feed occurs. When line 2 is closed, the charging valve is closed, and rapid traverse occurs.

Line 3 connects the spring end of the neutral-valve chamber with the pilot valve. Closing line 3 with the pilot valve causes the valve plunger in the neutral valve to seat. This is done by preventing the oil from escaping through line 3 to the pilot valve and back to the tank. When line 3 is closed, the neutral valve closes, and feed or rapid traverse occurs. When line 3 is open, the neutral valve is open, putting the circuit in neutral.

Line 4 is the slow-feed line. It connects the slow-feed-valve chamber with the pilot valve. The pilot valve connects line 4 with the rapid-traverse pump from which pressure (charging pressure) is obtained or opens it to the tank. The slow-feed valve is held in position for fast feed by a spring. Pressure in line 4 overcomes this spring and shifts the valve into position for slow feed. Opening the line to the tank allows the spring to force the valve to its former position, and fast feed takes place.

Line 5 connects the slow-feed-valve chamber with the feed adjustment housing.

Line 7 leads charging pressure from the rapid-traverse pump to the remote-control valve, which in turn distributes it to accomplish various functions.

Operation of Sundstrand Circuit. The constant-displacement pump, which is self-priming, pumps oil to a chamber open to the valve side of both the neutral valve and the charging valve. Opening and closing these two valves controls both the constant- and variable-displacement pumps. A schematic layout of the circuit is shown in Fig. 196.

Rapid traverse takes all the oil from the constant-displacement pump, forcing it through the piston chambers in the piston pump and on to the main cylinder. This is accomplished by actuating the pilot valve, which closes the line 2 as well as line 3, thus closing both the neutral and charging valves. The direction of rapid traverse is determined by the position of the four-way valve stem. In the 5PWX and 10PWX pumps, the four-way valve is so constructed that when put in reverse rapid traverse, line 2 is automatically blocked off, which eliminates a reverse feed.

Feed, either fast or slow, is obtained by opening the charging valve, which establishes sufficient pressure to keep the pistons in the piston pump up against the wobble plate, thus charging the piston pump, which is not self-priming. The excess oil from the constant-displacement pump returns through the charging valve to the tank. In fast feed, lines 2 and 4 are open to the tank, and line 3 is closed. In slow feed, line 2 is open to the tank; line 3 is closed, and line 4 is open to the charging pressure. This pressure shifts the slow-feed-valve plunger, opening pressure in line 5 from the piston pump to the wobbler-support plunger in the piston pump, which is forcing the plunger forward to provide the slow-feed rate by decreasing the piston travel.

THIRD FEED RATE. The third, or intermediate, feed rate is obtained with a special feed-adjustment housing containing an auxiliary wobbler-plunger feed-adjusting screw, and feed-adjusting cam, used in conjunction with the solenoid-operated-valve control. An additional solenoid valve is added and is connected to a high-pressure port in the pump housing and to a port in the feed-adjustment housing. When the solenoid valve

is energized, pressure from the high-pressure port in the pump is admitted to the feed-adjusting housing and acts on the auxiliary wobbler plunger. Deenergizing the solenoid valve blocks the high-pressure port and opens the port in the feed-adjustment housing to the tank. The feed-adjusting mechanism is then free to be shifted to one of the other rates.

CHARGING PRESSURE. When starting the moving member of a machine and in feed, the charging pressure is about 60 psi. In rapid traverse, the charging pressure is the same as the working pressure, which pressure is determined by whatever force it takes to actuate the moving machine member at the rapid-traverse rate.

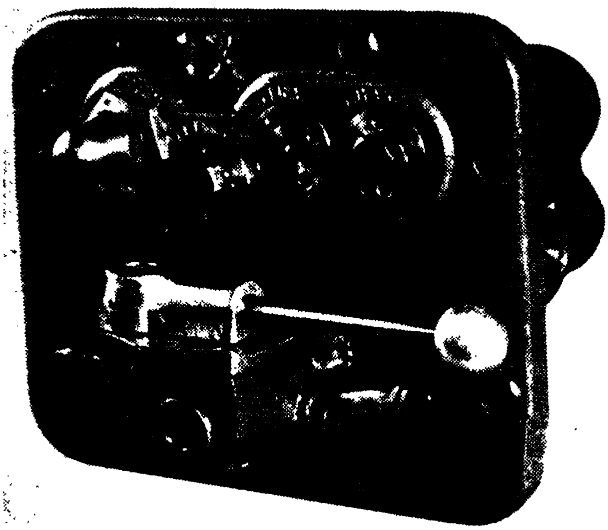


FIG. 198. The 1-X control valve. (*The Sundstrand Machine Tool Co., Rockford, Ill.*)

FEEDING PRESSURE. The operating or working pressure in feed depends entirely upon the force required to do the work. There is a minimum working pressure established by the back-pressure valve.

CYCLES OBTAINED. 1. Rapid traverse forward, either one or two rates of feed, and rapid return are obtained with following units:

5PWX or 10PWX pump and 14X control valve
5PWX or 10PWX pump and 101X triple solenoid valve

2. Rapid traverse and either one or two rates of feed in both directions:

6PWX or 11PWX pump and 16X control valve
6PWX or 11PWX pump and 102X triple solenoid valve

In addition to the PWX pumping unit with integral valving, a plain pumping unit may be supplied in connection with a complete control panel, containing all control valves shown in Fig. 196 plus a built-in pilot valve that may be actuated manually or by trip dogs from the machine-tool table. This panel unit is illustrated in Fig. 198.

The Oilgear Co. The Type F Oilgear Fluid Power Feed. This ingenious device, shown in exterior view in Fig. 199, consists of a constant-

delivery rapid-traverse pump, a variable-delivery feed pump, and a directional-control valve, all assembled in a compact casing together with relief and back-pressure valves. The casing may be flange mounted to a machine-tool housing, or it may be mounted separately in an oil pot. Several models are available, having rapid forward speed, one or two feeding speeds, and rapid return. One or two feeds on return stroke are also available. An optional delayed-reverse mechanism may be supplied,

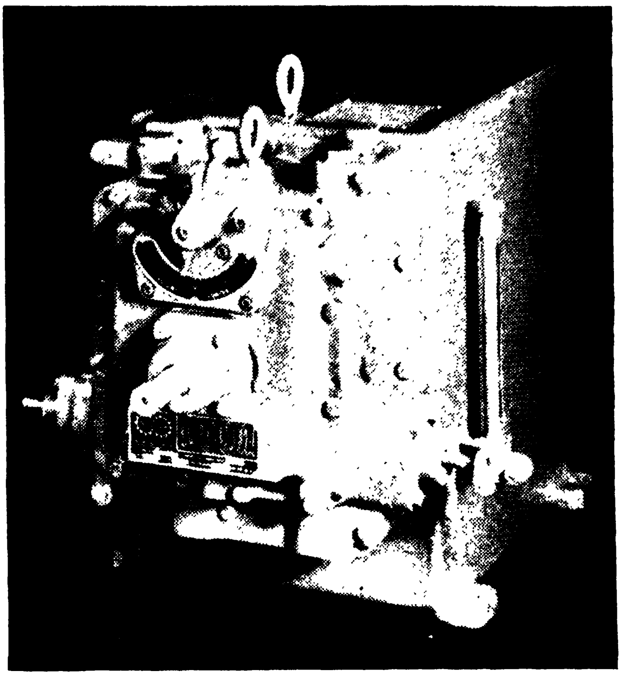


FIG. 199. Type F fluid power feed. (*The Oilgear Co., Milwaukee, Wis.*)

and a remote-control pilot cylinder is available to provide stopping or starting from a remote point.

At 860 rpm, feed-pump capacity is 260 cu in. per min, and two options of rapid-traverse capacity are available having 2,600 and 5,200 cu in. per min, for 3- or 5½-hp drive. At 1,150 rpm, feed capacity of 350 cu in. per min is available with 3,500 and 7,000 cu in. per min rapid-traverse capacity, corresponding to 4- and 7½-hp drives. The unit has a maximum feed pressure of 1,000 psi and rapid traverse of 300 psi. The unit weighs 250 lb without oil pot. Standard oil-pot capacity is 6 gal. Control of the unit is accomplished by a hand lever or foot pedal, attached to the control-valve device by suitable linkage, so that the control valve may be positioned in any one of its locations to produce the several speeds and directions of control. For automatic operation, dogs on the machine-tool table may be provided to shift the control valve to its different positions, with a load-and-fire mechanism to get the control past its neutral or stop position. The manufacturer will provide drawings and

designs for suitable mechanism to do this. The company has also brought out an electrical control attachment, consisting of hydraulic pilot-operated plungers, actuated by electrically controlled pilot valves, which are compactly mounted to the feed unit and may be remotely controlled by push buttons and limit switches, thus dispensing with all linkage mechanism and permitting mounting the hydraulic feed unit

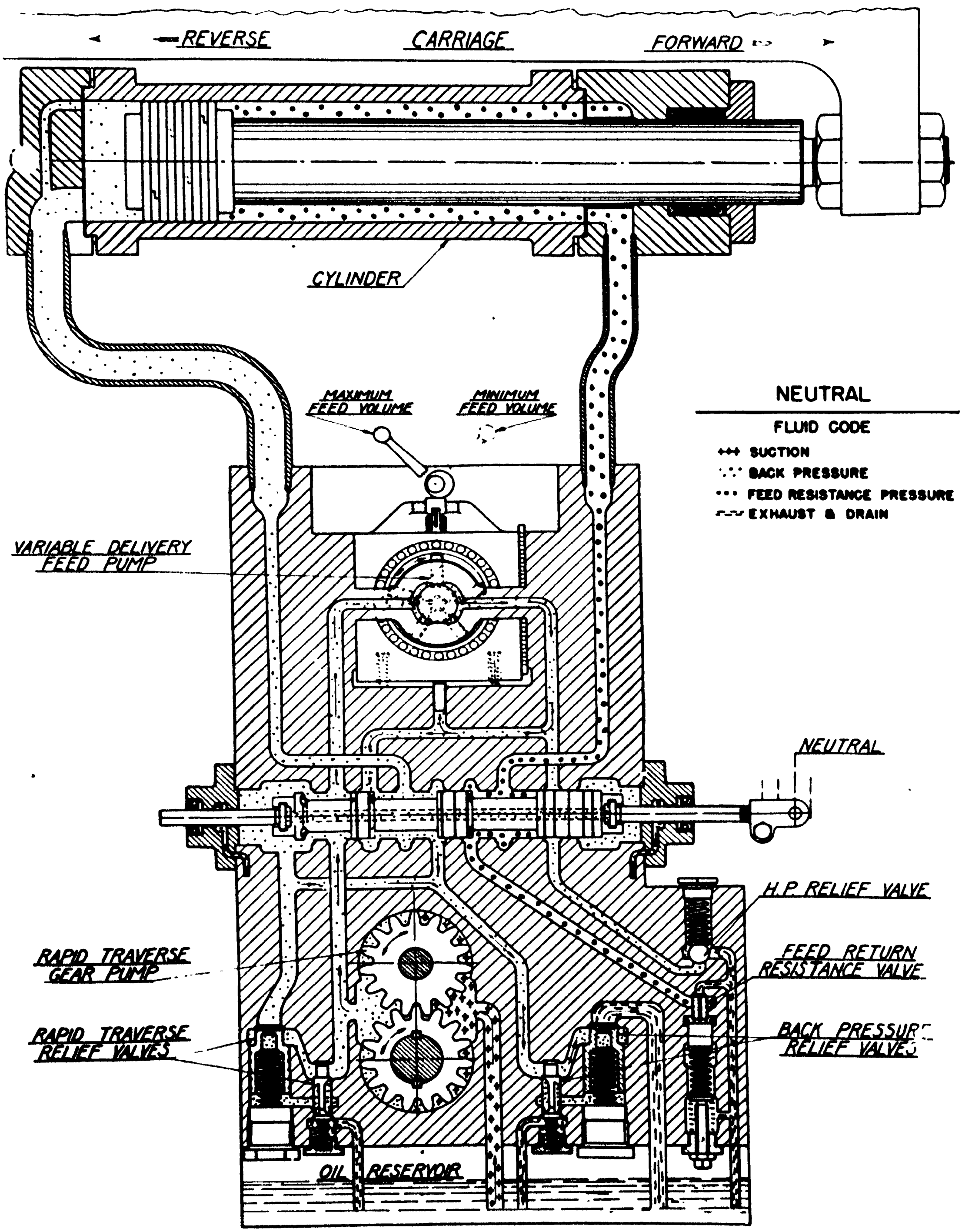


FIG. 200. Oilgear fluid power feed, neutral position. (*The Oilgear Co., Milwaukee, Wis.*)

remotely or concealed, without mechanical connection to the actuated members.

The diagrammatic sketches (Figs. 200 to 203) illustrate the components of the unit and their operation in the different positions of the control valve. Figure 200 illustrates the unit in neutral position. In this posi-

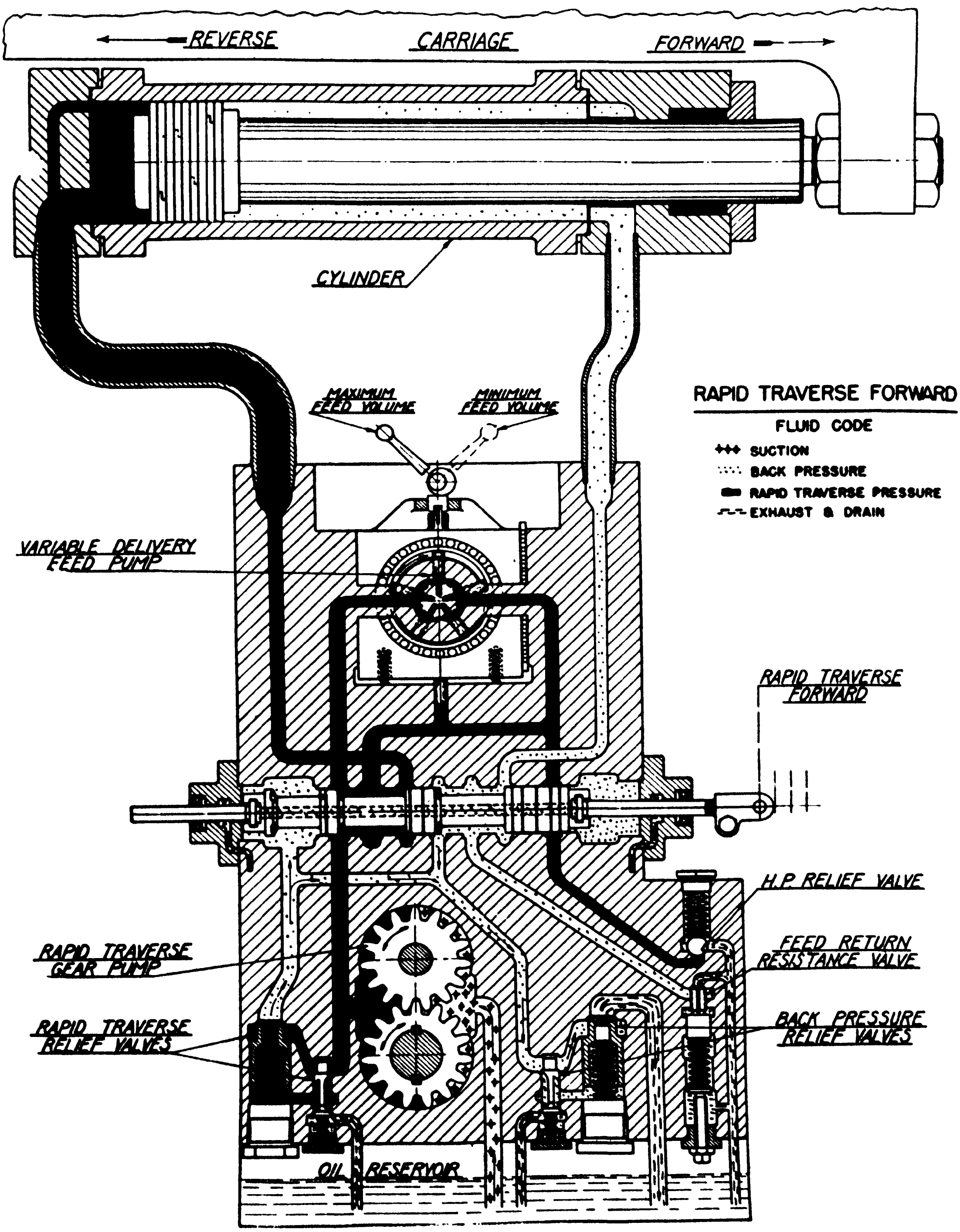


FIG. 201. Oilgear fluid power feed, rapid traverse forward. (*The Oilgear Co., Milwaukee, Wis.*)

tion, the rapid-traverse pump discharges fluid under back pressure (5 to 15 psi) to the forward end of the machine cylinder, simultaneously supercharging the feed pump, the output of which joins the output from the rapid-traverse pump at back pressure. The back pressure is produced by the back-pressure relief valve, a pilot-actuated relief valve operated by a small pilot valve. The machine cylinder is locked and prevented from moving by the feed-return resistance valve, set at about 100 psi.

Rapid Traverse Forward (*Fig.* 201). With the control valve shifted to the rapid-traverse-forward position, oil is discharged to the forward end of the feed cylinder at rapid-traverse volume and pressure up to 300 psi, determined by the setting of the rapid-traverse relief valve. Oil also passes through the feed pump. The return end of the feed cylinder is open to back pressure. This back pressure may not be sufficient to support the weight of a vertical moving slide, which should be counterbalanced to prevent uncontrolled drop.

Feed. Figure 202 shows the feed position of the unit. The rapid-traverse pump is supercharging the feed pump at back pressure. The feed pump discharges into the forward end of the feed cylinder at a rate determined by the setting of the feed-adjustment cam. Oil discharged by the return end of the feed cylinder escapes through the resistance valve, thus locking the feed piston between fluid columns. The diagram also illustrates the leakage compensator, which maintains a constant feeding speed regardless of pressure. To this end, the feed-adjusting cam operates upon the feed-pump slide block through a set of Belville washers. Maximum volume is reached with the cam retracted, so that the opposing springs may force the pump slide block into maximum-discharge position. Reduction of output is brought about by rotating the adjusting cam, so as to force the pump slide block toward neutral position. An auxiliary plunger is provided, which assists the action of the opposing springs. As pressure builds up in the feed line, leakage in the pump tends to increase, which would result in a decrease of output, were it not for the action of the auxiliary plunger. The plunger forces the slide block on more stroke, compressing the Belville washers, which are so calibrated that the increased stroke, owing to the action of the auxiliary plunger, will compensate for the additional leakage due to the increased feed pressure.

Rapid Return (*Fig.* 203). In this position, oil is discharged from the rapid-traverse pump to the retraction end of the feed cylinder. The forward end of the feed cylinder is at back pressure. Thus the tool slide is returned at rapid-traverse speed to its initial or starting position.

The new Type JK power feed, recently developed by the Oilgear Co., has characteristics similar to the Type F power feed, but possesses several

unique characteristics. The unit is completely electrically controlled by means of push-type solenoids that act directly on two selector control valves without intervening pilot valves. Rapid forward, two feeding speeds, and rapid return are provided. The pressure-compensated feed pump has a flat-plate valve instead of the conventional pintle.

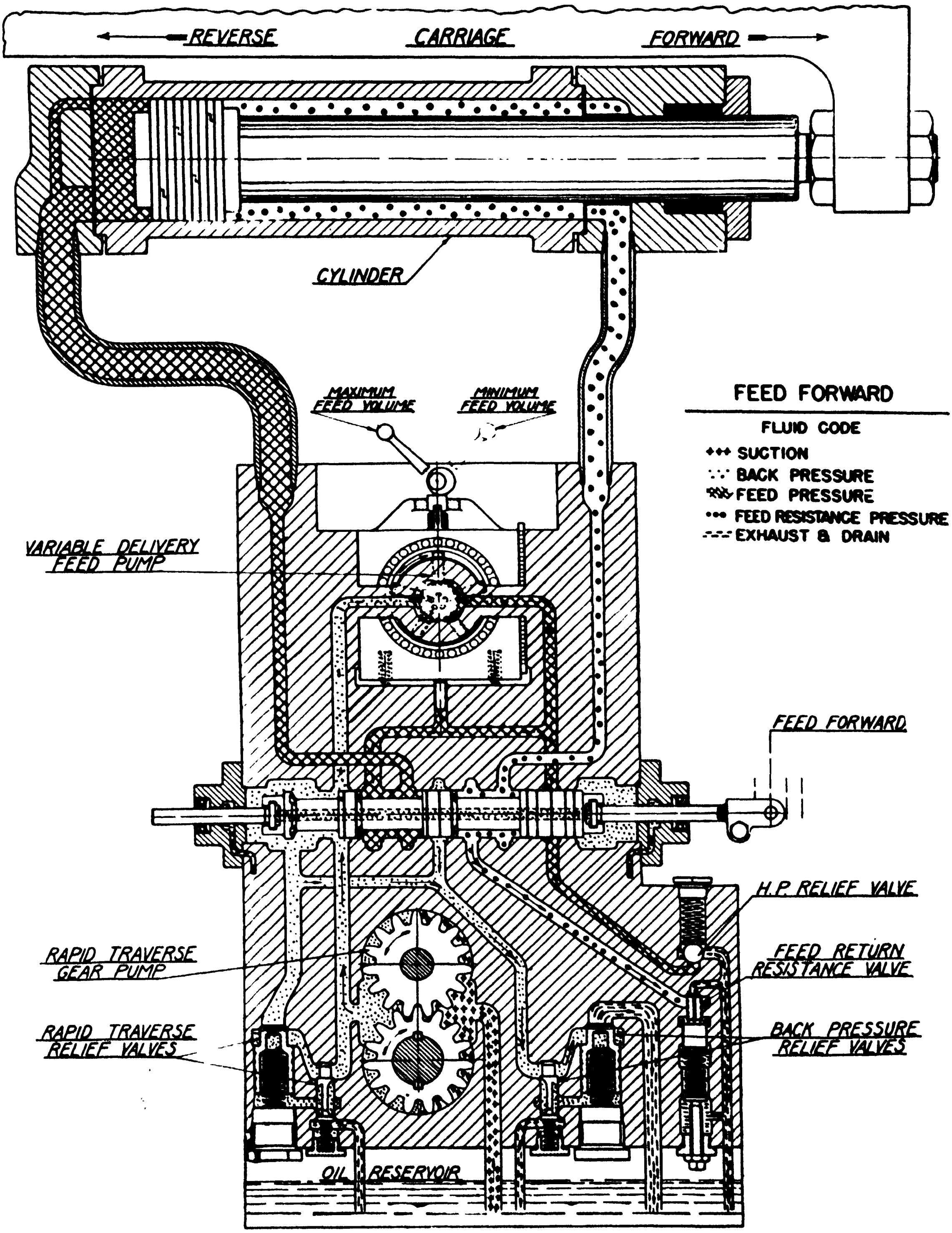


FIG. 202. Oilgear fluid power feed, feed. (*The Oilgear Co., Milwaukee, Wis.*)

Feed-pump delivery at 1,200 rpm is 150 cu in. per min maximum to 10 cu in. per min minimum, and rapid-traverse delivery at the same speed is 2,100 cu in. per min. A 2-hp motor is required to operate the unit at that speed and the recommended pressure of 300 psi rapid-traverse and 1,000 psi feeding pressure.

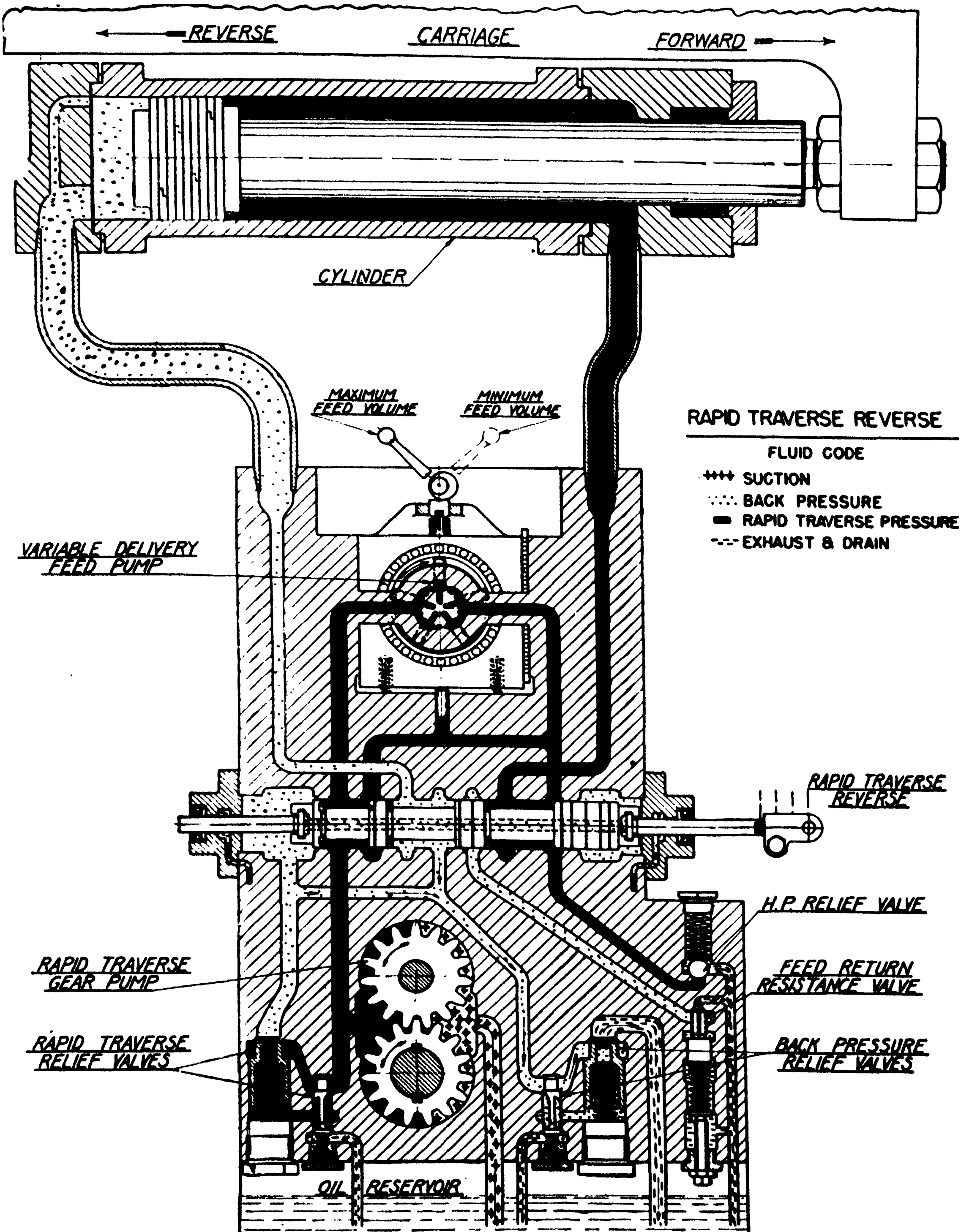


Fig. 203. Oilgear fluid power feed, rapid return. (*The Oilgear Co., Milwaukee, Wis.*)

6. STORAGE AND INTENSIFICATION OF HYDRAULIC POWER

Storage of hydraulic power is one of the oldest applications of such power. Accumulators, consisting of cylinders and rams, loaded with cast iron or concrete, were a familiar picture in steel mills, pressing shops, and other places where hydraulic power was used to any extent. Development of oil-pressure hydraulics, and particularly the invention of the variable-delivery pump, brought on a gradual decline of the accumulator system, and oil-pressure hydraulics proceeded at an accelerated pace of development without these cumbersome and hazardous appliances. Gradually, a modern system of hydraulic-pressure storage began to make its appearance. This is storage of hydraulic power by elastic-fluid-loaded storage vessels. Development of water-pressure storage vessels has paralleled this development, and huge vessels for storage of water pressure for extrusion machines, forging presses, etc., have been built and installed. In a similar manner, oil hydraulic storage vessels are being used for die-casting machines, welders, and other applications that require large amounts of power for short intervals, spaced so that the fluid withdrawn may be made up by the pump in between these intervals.

While in water-pressure storage vessels, the water is in direct contact with the elastic pressure medium, it is much preferable to separate the two mediums in an oil-pressure system. To this end, pressure vessels with moving pistons and elastic diaphragms have been developed. Figure 105 shows such a device in connection with an Oilgear pump and pressure-compensating control. The design of these pressure vessels follows closely that of regular hydraulic cylinders. Generally the vessels are made from steel forgings and have a floating piston in a smooth bore. The stroke of the piston is limited by a stop or shoulder, so that the compression of the elastic medium (generally nitrogen) is limited. The displacement of the piston determines the capacity that may be withdrawn from the unit and bears a relationship to the total volume of the vessel, determined by Boyle's law (see Sec. 2, Chap. II). For a full understanding of the operation of these accumulators, this relationship must be borne in mind, particularly the fact that every change in volume of the accumulator, such as withdrawal or addition of fluid, is accompanied by a change in pressure. The total amount of pressure fluctuation may be determined by the designer in dimensioning the relative values of piston displacement and total volume and is generally made from 10 to 20 per cent. Thus the total volume should be from six to eleven times the piston displacement. For instance, if the piston displacement is 100 cu in., then the total available volume in the bottle

should be 600 cu in. When the piston is then pumped back and the vessel charged with 100 cu in. of oil, the pressure will increase to six-fifths of its original value, or gain 20 per cent. This, and the fact that the pistons in these bottles have a friction factor of about 15 per cent total (difference between charging and discharging pressure), dictates great caution in properly selecting pressure capacity and horsepower of pumps. In the above case, for instance, if a minimum system pressure of 1,000 psi is required, the pump must deliver 1,000 × 1.15 psi at the start and 1,000 × 1.15 × 1.20 psi at the end of the charging operation. The pump and motor must, therefore, be capable of supplying 1,380 psi, so that the system will be assured of receiving the minimum of 1,000 psi. This piston-type accumulator is well adapted for large volumes and heavy pressures in connection with variable-delivery pumps and pressure-compensating controls. The vessels may be charged with nitrogen from commercial bottles and should be equipped with suitable connections and a Kerotest valve or equivalent with fusible plug that melts in case of fire and permits the gas to escape.

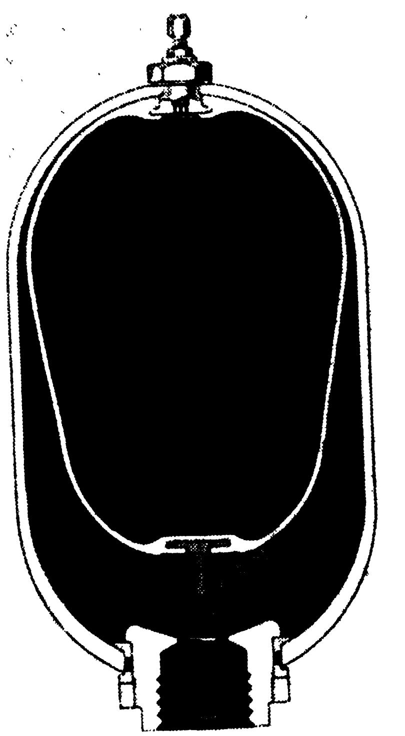


FIG. 204. Bladder-type hydraulic accumulator. (*Greer Hydraulics Inc., Brooklyn, N.Y.*)

For smaller capacities, accumulators of the diaphragm or bladder type are suitable. The diaphragm-type accumulator is made from two hemispheres, separated by a flexible diaphragm, with elastic fluid, such as air or nitrogen, in one end and the oil in the other. The capacity-pressure relation in these accumulators is the same as in a piston-type unit, following Boyle's law. Vickers Inc. supplies a unit having a total volume of 69 cu in., good for 1,500 psi maximum pressure. If the unit is charged with a given pressure with no oil in the oil end, then pumping about 34 cu in. of oil into it will double the pressure, 46 cu in. will triple it, etc. If we want to hold the pressure variation in this particular unit to 20 per cent of the charging pressure, we must not add or withdraw more than 11½ cu in. A bladder type of pressure vessel is shown in Fig. 204. This unit, made by Greer Hydraulics Inc., is available in several sizes and pressure capacities up to 10 gal capacity, and 3,000 psi. Its pressure-volume characteristics are, of course, the same as those of the other types.

Since diaphragm and bladder accumulators have no pistons, there is no piston-friction factor to consider; nevertheless the efficiency is not quite 100 per cent, owing to the work of expanding and compressing diaphragms and bladders. The units may be used in connection with variable-delivery pumps and pressure-compensating controls or with constant-delivery pumps with suitable unloading valves. These unloading valves, an outcome of aircraft hydraulic design, are made by Vickers and others and consist in principle of pressure-controlled pilot-actuated by-pass valves that permit the pump to idle at substantially zero pressure when the pressure in the accumulator has reached a predetermined setting.

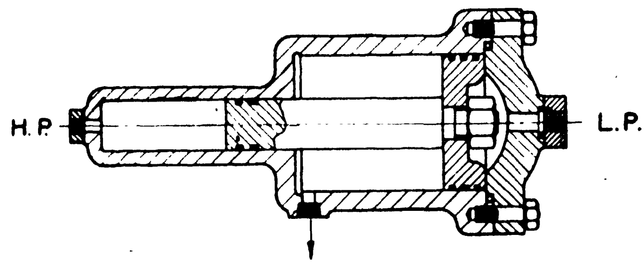


FIG. 205. "One-shot" intensifier.

Pressure drop, due to discharge of the accumulator, will cut the pump back in, so that it will again charge the accumulator.

Intensification of hydraulic pressure is an old and well-known hydraulic practice. Its use and development dates back to the use of accumulator systems. In the operation of heavy hydraulic equipment, such as forging presses, keel benders, etc., the desirability of going to high hydraulic pressures became quite apparent, owing to the saving in weight and bulk that could be accomplished. Generation and storage of these pressures, however, proved to be impractical, because of the heavy maintainance expense, costly equipment, etc. So the compromise was evolved to generate hydraulic power at moderate pressures, say 3,000 to 5,000 psi, and then to raise it to higher values of up to 15,000 psi by means of intensifiers. The gradual decline of water hydraulics, taking place prior to the introduction of oil hydraulics, caused a temporary loss of interest in these devices, which seemed to have little value for oil hydraulic applications, mainly because the intensifier then known was only a "one-shot" device, limited in its capacity, and had to be recharged after each operation. A single "one-shot" intensifier is merely a hookup of two rams of different diameters, the cylinders of which are tied together with tie rods or other means, so that the action of one ram opposes that of the other. If hydraulic pressure of a given magnitude is supplied to the larger one of the two rams, then obviously the smaller one will produce a pressure equal to the supply pressure times the ratio of areas of the two

rams, times the efficiency of the device, which in general runs from 90 to 95 per cent. Modernized versions of this device are still made, in design similar to Fig. 205. This rather simple device permits application of high pressure, the capacity being limited by the displacement of the high-pressure plunger. After each operation, the plunger must be retracted and the unit recharged.

The apparent disadvantages of the "one-shot" system led to the development of continuous intensifiers, a well-known example of which

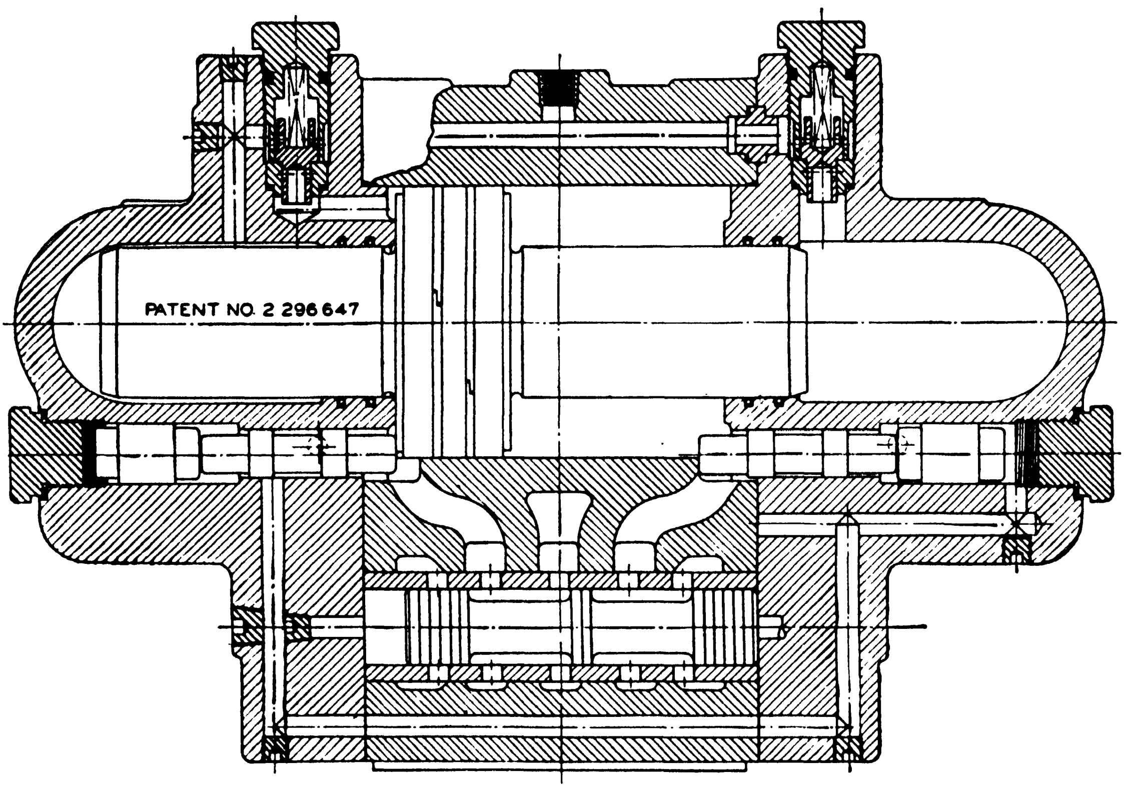


Fig. 206. Hydraulic pressure booster, sectional diagram. (*Racine Tool and Mfg. Co., Racine, Wis.*)

is the Racine pressure booster. A diagrammatic section of this ingenious device is shown in Fig. 206. In principle the device consists of a differential double-acting piston that produces intensified pressure at the small ends and is operated by low pressure acting on the combined areas on the opposite ends, so that the intensification ratio is the ratio between the squares of the large central and the small end rod diameters. Application of low or driving pressure is controlled by a pilot-operated four-way valve, which admits low pressure to either one or the other differential area and from there through a suitable check valve to the small plunger area on the same side. This forces fluid from the opposite small plunger through the high-pressure check valve into the high-pressure discharge line, while the differential area on this opposite side is connected to the

exhaust by the same pilot-operated four-way valve. Two three-way pilot valves, actuated by the intensifier motor piston at each end of its stroke, control the operation of the four-way valve and are interlocked hydraulically, whereby the action of one pilot valve, when operated by the motor piston, not only supplies pressure for shifting the main four-way valve, but also for shifting the opposite pilot valve, so as to relieve pressure on the opposite end of the four-way valve and release the hydraulic interlock on the valve just shifted. The opposite pilot valve

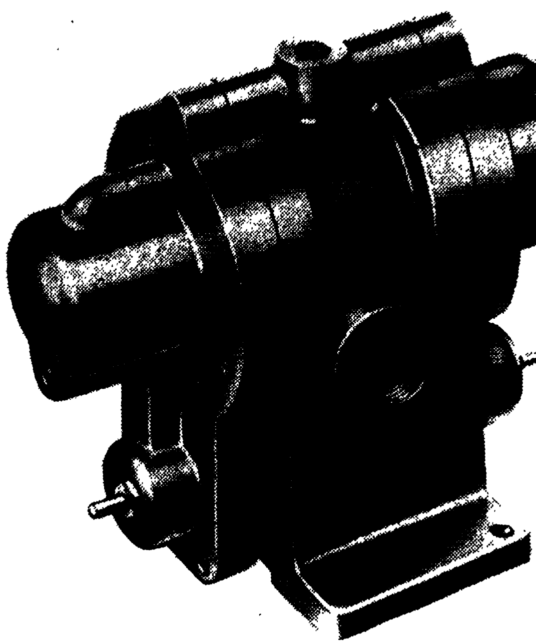


FIG. 207. Hydraulic pressure booster. (*Racine Tool and Mfg. Co., Racine, Wis.*)

remains hydraulically locked until operated by the motor piston on its return trip. In connection with the Racine variable-delivery vane pump, the Racine booster has found numerous applications for supplying and holding high hydraulic pressures without the necessity for supplying a costly high-pressure pump. Racine boosters are available for low-pressure capacities up to 24 gpm and a number of pressure ratios from 3:1 to 7:1. Maximum high pressure for standard units is 3,000 psi. An exterior view of the unit is shown in Fig. 207.

The Hydro-Power Hydraulic Booster. This continuous intensifier, illustrated in Fig. 208, is a multiple-pressure unit providing smooth and uninterrupted flow of high-pressure fluid. A rotating central distributing valve directs the flow of the low-pressure pump successively to the combined area of the plungers, while the small area on the opposite end is connected to the high-pressure outlet and its differential area to the

exhaust. There is an even number of plungers, and diametrically opposite plungers are always connected together by the central valve, so that the plungers operate in pairs, and the central valve is completely pressure balanced. Each piston makes two complete strokes per revolu-

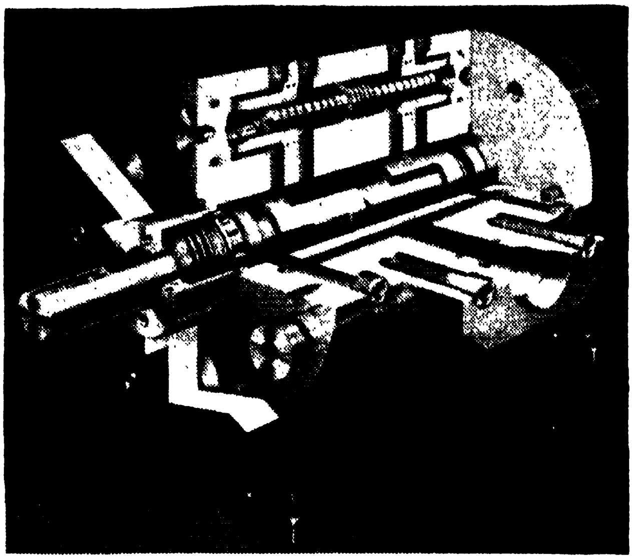


FIG. 208. Hydraulic booster. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

tion. The unit is available in two ratios, 2:1 and 3:1, and two sizes, 35 and 100 gpm input. Maximum output pressure is 7,500 psi. The central valve must be rotated either by a separate small motor or by take-off from the low-pressure-pump driving motor.

CHAPTER XI

THE APPLICATION OF OIL HYDRAULIC POWER

1. General Considerations. In preceding chapters we have dealt with the components that go to make up a hydraulic system: pumps to generate hydraulic power, transmission lines that convey the power to the point of utilization, control devices that direct the application of hydraulic power and control flow and pressure, and hydraulic motors of rotary or reciprocating type that convert hydraulic energy into useful work at the point of application. This chapter will deal with the coordination of these components into a useful working mechanism, in which primary power, generally electric current, is put to work through a series of transformations to operate production tools, lift or position heavy weights, control simple and intricate motions, in short, to perform innumerable useful services in production processes.

The engineer who is entrusted with the responsibility of designing a machine, tool, or mechanism, will have to make a decision on what mode of power transmission is indicated, and in numerous cases application of hydraulic power will be found advantageous and useful. To arrive at this decision, he will do well to provide himself with a check list enumerating the possible uses of hydraulic power, which may be compared with his own problem. In many cases the use of hydraulic power is mandatory or strongly suggests itself; in others a decision is less easily arrived at. In recent advertising, the Oilgear Co., Milwaukee, has published such a list, which is reproduced here with their permission. The profitable use of hydraulic power is indicated if one tries to

1. Apply large forces through long or short strokes at variable speeds
2. Obtain automatic work cycles, variable speeds in either direction, with or without preset time dwell
3. Apply large forces through continuous or intermittent reciprocating cycles at constant or variable velocities
4. Obtain extremely accurate control of either speed or position of a reciprocating member
5. Apply accurate variable pressure, either static or in motion
6. Closely synchronize various motions, operations or functions
7. Apply light, or heavy, forces at extremely high velocities through either long or short distances of travel

8. Obtain continuous automatic reversing drives at constant rpm or over a wide range of speed variations

9. Obtain accurate remote control of speed and direction of rotation, rates of acceleration and/or deceleration

10. Obtain constant-horsepower output through all or part of a speed range

11. Obtain automatic torque control

12. Obtain accurately matched speed of various rotating elements

13. Obtain constant-speed output from a variable-speed input

14. Obtain full preset automatic control, elimination of problems of shock, vibration, etc.

After the decision has been made to adopt fluid power for the operation of the contemplated machine, an engineering study must be made to establish the type of hydraulic system best suited for that particular application. In particular, this refers to low or medium pressure vs. high pressure, variable delivery vs. constant delivery, accumulator vs. direct pumping.

To set up hard and fast rules for the decision is not advisable, as specific cases must be dealt with on their individual merits, but the following general information will be found useful. Pressures of approximately 1,000 psi or less are recommended

1. Where close control and accurate positioning are more important than brute force

2. Where cylinder and ram rigidity are essential

3. Where hydraulic power is transmitted over relatively short distances

4. Where horsepowers are relatively low, approximately 25 hp or less

5. Where cost limitations indicate the employment of lower cost low-pressure rotary pumps, such as gear or vane pumps

On the other hand, employment of pressures in the neighborhood of 3,000 psi or more is indicated

1. Where large forces must be exerted, particularly at high speeds

2. Where it is necessary to limit power losses due to long transmission lines

3. Where horsepower requirements are high

4. Where reduction in cylinder, machine, and valve and piping size, due to employment of higher pressures, outweighs the greater cost of high-pressure pumps

As we have seen, most low- and medium-pressure pumps, with few exceptions, are of constant-delivery type. On the other hand, most high-pressure pumps now on the market are of the variable-delivery and reversible-discharge type. Thus, decision between high- and low-pressure system to some extent determines the question of constant vs.

variable delivery. This is not entirely true, as employment of the variable-delivery principle is often indicated on low-pressure jobs, particularly where saving of power and high efficiency at partial loads is desired and/or where pump is required to maintain a deadhead pressure in which a high- and low-pressure pump combination is a poor second choice compared to a variable-delivery pump with pressure-compensating control. On the other hand, a successful constant-delivery high-pressure installation may be made in cases where the equipment is called upon to operate at its highest speed practically continuously, and where deadhead or holding pressures are not required. Consideration of economy enters again, as variable-delivery pumps are more expensive, and compromise solutions must sometimes be made where cost is an important factor.

By far the majority of hydraulic systems operate with one-way pumps, both of constant and variable delivery, and directional control is accomplished by valving. There is a large class of applications, however, in which the employment of the reversible-discharge-pump principle holds outstanding advantages. This covers very heavy high-speed machines, such as oil hydraulic high-speed presses and similar heavy machinery. There the outstanding advantages of the reversible-discharge pump, which are shock-free reversal of heavy masses, controlled decompression of large volume, and smooth acceleration and deceleration, may be utilized to their fullest extent.

Most oil hydraulic systems are directly operated by pumps without accumulators. Development of bladder- and piston-type elastic-pressure-loaded accumulators is responsible for a good many recent installations employing this means of power storage. In cases where instantaneous demands of fluid power are high, followed by periods of low demand, great savings can be made in pump capacity and connected horsepower by the installation of an accumulator. This has been found advantageous for die-casting and forging machines.

Another important consideration is that of single pump vs. multiple pumps. Some applications have but one moving member that must be actuated hydraulically. This will then make a single pump-motor combination, where the motor may be of the reciprocating or rotating type. In many cases several moving members are involved, often leading to circuits of great complexity. A decision must then be made whether to provide individual pumps for these members or to operate them from one common source of hydraulic power. It may be said, in general, that low-pressure systems consisting of several members moving in sequence or simultaneously may be actuated from one single source of hydraulic power and controlled by suitable sequence and metering valves. Particular care must be taken to permit the pump to unload and by-pass at

substantially zero pressure at the completion of a cycle, and if this is done automatically by the last one of the moving members, provisions must not be forgotten to permit movement of some other member to start a new cycle.

High-pressure systems, in particular those which require deadhead or holding applications on some members, are generally made with multiple pumps. Thus it is customary on oil hydraulic presses, for instance, to operate the press proper with one pump and run auxiliaries, such as feeding tables, etc., from a separate unit.

Auxiliary so-called "pilot" hydraulic systems are found both with individual pilot pumps and with bleed-off from main system. Pilot pressure, when bled off from main system, is subject to system-pressure fluctuation and therefore movement of pilot-controlled valves can be controlled less accurately than if a separate pilot pump is provided. Some hydraulic systems provide for both sequence and reducing valve in the main system to hold bled-off pilot pressure constant. The sequence valve ensures maintainance of a minimum pressure, while the reducing valve limits the maximum pressure.

If after consideration of all factors a decision has been made on the type and pressure capacity of pump, and the question of single or multiple pumps has been decided, we may proceed to analyze the movements of the machine members to establish capacities and horsepowers and operating sequences. The size of the respective operating cylinders may, of course, be established from the known or estimated load they are to exert, once the operating pressure has been settled upon. In analyzing the time cycles for the different motions, it may often be found that material savings can be effected by provision of low-pressure or prefill pumps, since often parts of the cycle are performed at very low pressure, for example, the rapid-traverse motions on machine tools and presses. Pump capacities may be established by computing the displacement required from ram areas and ram speeds demanded by the time cycle. In computing these, consideration must be given to oil compressibility and other time-lag-producing factors, such as time required to shift controls, slowdown required before engaging stops, etc.

After establishing pump capacities, horsepower may be computed by taking into consideration the power-time relationship with due consideration for pull-out torques as shown in Sec. 5, Chap. VII.

A hydraulic circuit may then be roughed in, showing the different machine members and their connections with control and metering valves as determined by the cycle requirements, the pumps and pressure-control devices, and their hookup with the operating system. Actual dimensions of valves and piping are then determined to care for the flow requirements

of the different devices. So far as possible, standard valves and control devices should be used that may be obtained from manufacturers' stock. For automatic systems, time-cycle requirements determine the type of and arrangement of controls. To this end, mechanical devices such as cams, load-and-fire mechanisms, and levers may be used to provide automatic or semiautomatic control. Alternately, operation by hydraulic pilot system or electrical control may be preferred. For the latter an electric circuit diagram will have to be set up. Operation by push buttons, limit switches and solenoid-actuated valves is becoming increasingly popular. After the circuit has been set up, a careful check should be made to provide for interlocks, electric or hydraulic, to prevent damage to machine members from interfering with each other. The different cylinders should be checked to see whether they may be operated to the limits of their stroke without danger of damage by exceeding this stroke, and positive ram stops provided where necessary. Oil reservoirs should be checked for capacity so that they will permit operation of hydraulic cylinders without dangerously lowering the oil level. Attention should be given to elimination of air from the hydraulic system by provision of air cocks at high points of the system or automatic air bleeders. Dead pockets in which air may be trapped should be avoided.

The machine should then be checked not only for its normal operating cycle, but also for abnormal functions, such as emergency reverse, safety devices, etc. Finally a complete diagram may be drawn up, with consideration of the physical dimensions of the machine, to fit in the various devices, cylinders, and valves and run the connecting piping or tubing. The type and design of piping, fittings, etc., may then be determined. Full detailed information on any of the intermediate steps has been given in preceding chapters and may be referred to in following the outline given above for design of the hydraulic system.

In order to acquaint the reader with the methods used in setting up hydraulic circuits, the following section will be devoted to this important phase of hydraulic design.

2. Hydraulic Circuits. As pointed out by Hans Ernst,[1] development in hydraulics has paralleled that in the electrical field by following a double path—unit development and circuit development—and the functioning of an electric or hydraulic system depends as much on the particular arrangement of the circuit as on the detail construction of the associated units. In the following we will illustrate the development of typical hydraulic circuits, starting with simple and ending with more complex systems. In these circuits, standard hydraulic units or com-

[1] Hans Ernst, Modern Hydraulic Control System and Circuits, *Product Eng.*, 121, April, 1935.

ponents made by different manufacturers will be shown diagrammatically, and for their functioning reference may be made to Chap. X, in which these control components have been illustrated and described.

Figure 209 shows an elementary circuit for operation of a double-acting cylinder. Oil discharged by the pump is directed to either side of the cylinder by a directional-control valve. This valve should be of the center–by-pass type to avoid shock when passing the neutral position. The pump may have constant delivery or variable delivery with a suitable

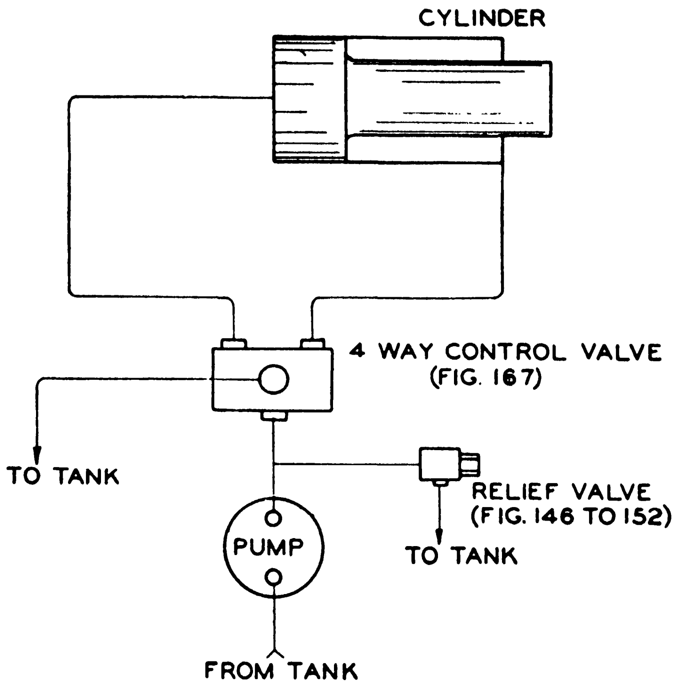


FIG. 209. Elementary hydraulic circuit.

control. A relief valve protects pump and system against overload. The relief valve may be of any of the constructions shown in Figs. 146 to 152, Chap. X. The valve may be manually operated and preferably spring centered so that pump will by-pass freely when valve is in neutral position, or it may be actuated by any of the automatic means that will be described later.

If we assume now that the cylinder is mounted vertically, rather than horizontally, the problem arises to maintain the piston suspended while the pump is idling. This may be done by installation of a back-pressure valve, as shown in Fig. 210. This valve, set slightly in excess of the pressure required to sustain the moving weight, prevents gravity drop and maintains the piston suspended in the by-pass position of the operating valve.

In Figs. 209 and 210, the area for retraction of the piston is smaller than the working area. This means that on retraction stroke the amount of oil

discharged from the head end is larger than the pump output by the ratio of head area to retraction area, and the operating valve and piping must be of sufficient size to accommodate this. In cases of very large differentials, this would lead to unduly large and expensive operating valves, and it may be found preferable to install an unloading valve in the head-

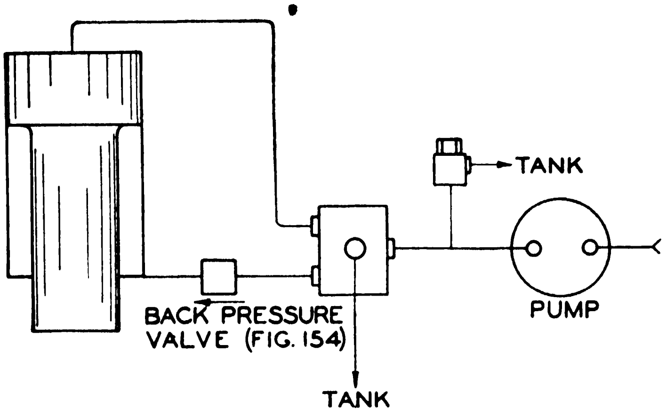


FIG. 210. Hydraulic circuit with back-pressure valve.

end line, as shown in Fig. 211. With this arrangement a high-speed retraction stroke of the piston is possible.

In many cases, a high-speed advance of the piston is desired, because very often only a fraction of the work stroke has to be performed under pressure. In this case, several solutions present themselves. Commonly

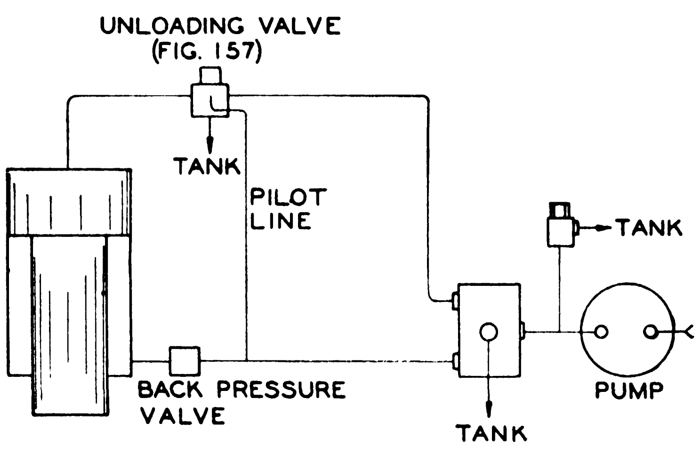


FIG. 211. Hydraulic circuit with unloading valve.

a two-pump system is used for this purpose, having a large low-pressure rapid-traverse pump combined with a small-capacity high-pressure pump, so arranged that at the maximum low pressure an unloading valve is opened, unloading the full capacity of the low-pressure pump. A check valve must be provided to permit the high-pressure pump to build up the pressure without backing into the low-pressure line. This arrangement is shown in Fig. 212. All other valves previously shown are retained, so that we now have a vertical piston operated by a high- and low-pressure

combination pump for rapid-traverse closing stroke, followed by a slower working stroke upon building up of a predetermined pressure, also rapid-traverse opening stroke. Accidental dropping of the piston by gravity is prevented by the back-pressure valve. Excess oil on the return stroke is discharged through the unloading valve. The ram will stop when the main operating valve is put in neutral position.

The size of the operating valve is based on the combined capacity of both pumps. The size of the head-end unloading valve is determined by the following consideration. If the assumption is made that on the

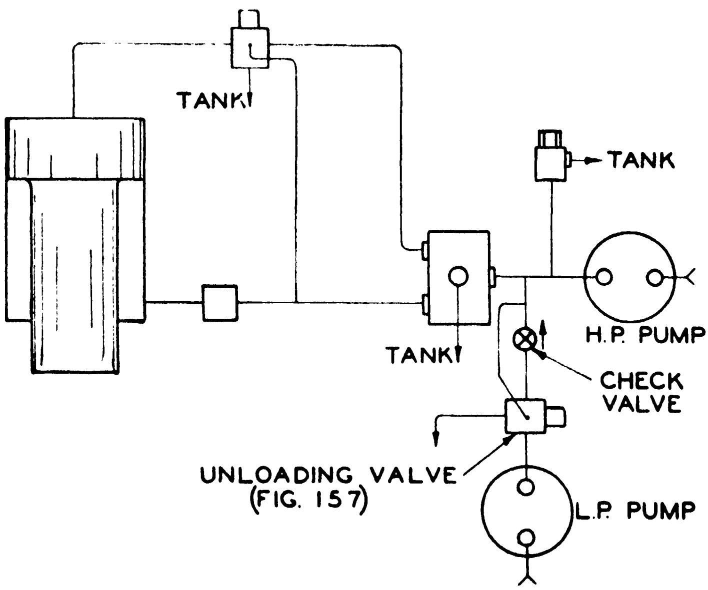


FIG. 212. High- and low-pressure operating system.

retraction stroke only the high-pressure pump is used and the pullback pressure is sufficient to unload the low-pressure pump, then the head end of the piston will discharge an amount of oil equal to the displacement of the high-pressure pump times the ratio of head-end and retraction area. If this amount is less than or equals the combined capacity of both pumps, no unloader is needed. If it is more, then an unloader must be installed of sufficient size to care for the difference. Maximum economy may therefore be achieved by making the ratio in areas equal to the combined capacities of both pumps divided by the high-pressure capacity. If it is desired to have both pumps participate in the retraction stroke, an unloader of sufficient size must be installed to care for the combined discharge of both pumps multiplied by the ratio of areas minus one.

Vickers high- and low-pressure units come mounted in one housing (see Fig. 67) equipped with integral check, unloading, and relief valves (see Fig. 160). A great variety of high- and low-pressure combinations is available to meet almost any requirement. As a special case of pump

combination, the pressure on both pumps is sometimes made nearly alike and the capacity of one of the pumps very small, so that at the end of a working stroke, a dwell may be maintained on the work by the small pump, while the large pump is unloaded. Power loss and heating are thus restricted to the capacity of the small pump. There is nothing to prevent making the small-capacity high-pressure pump of variable-delivery type, so that an adjustable pressure may be held with the larger pump by-passing.

In some applications a high-speed advance of the piston is desired, followed by a slower working travel to be established by the position of the traveling piston rather than by pressure. In that case, the pressure-actuated unloading valve in Fig. 212 may be replaced by a cam-and-roller-operated three-way valve that is engaged by the piston at any desired point of its working stroke.

If very high speeds of advance and retraction are required, both valves and auxiliary pumps become unwieldy, and a different means of obtaining this result should be employed. To this end, a so-called "prefill," or "surge," system is used. Rather than oil being discharged into the cylinder by means of an auxiliary pump during the rapid-advance stroke, the piston is advanced either by gravity or auxiliary rams, while the cylinder prefills through a specially designed valve from a prefill or surge tank. A circuit incorporating this arrangement is shown in Fig. 213. Booster or kicker cylinders are used to push a platen and large ram toward the work. A four-way directional-control valve, manually operated, directs pump discharge into forward or retraction area of the auxiliary cylinders. A counterbalance or back-pressure valve prevents gravity drop of the moving weights. While the platen descends under the influence of the pressure exerted by the booster rams, the cylinder fills with oil through the large prefill valve mounted between it and the oil-supply tank. When resistance is encountered by the platen exceeding the capacity of the cylinders, pressure conducted through a branch line operates a sequence valve built into the prefill valve and causes the latter to shut off the flow of oil from the tank, at the same time admitting pressure flow into the main cylinder. Reversing the directional-control valve causes pressure to be applied to the retraction area of the booster cylinder; simultaneously, a branch line applies pressure to a retraction piston on the prefill valve and forces it open, so that the platen may return at high speed, discharging oil from the main cylinder back into the tank. The circuit in Fig. 213 shows a downward-acting ram. By an obvious rearrangement, an upward-acting ram may be operated in the same manner.

Where moving weights attached to the hydraulic ram are of sufficient

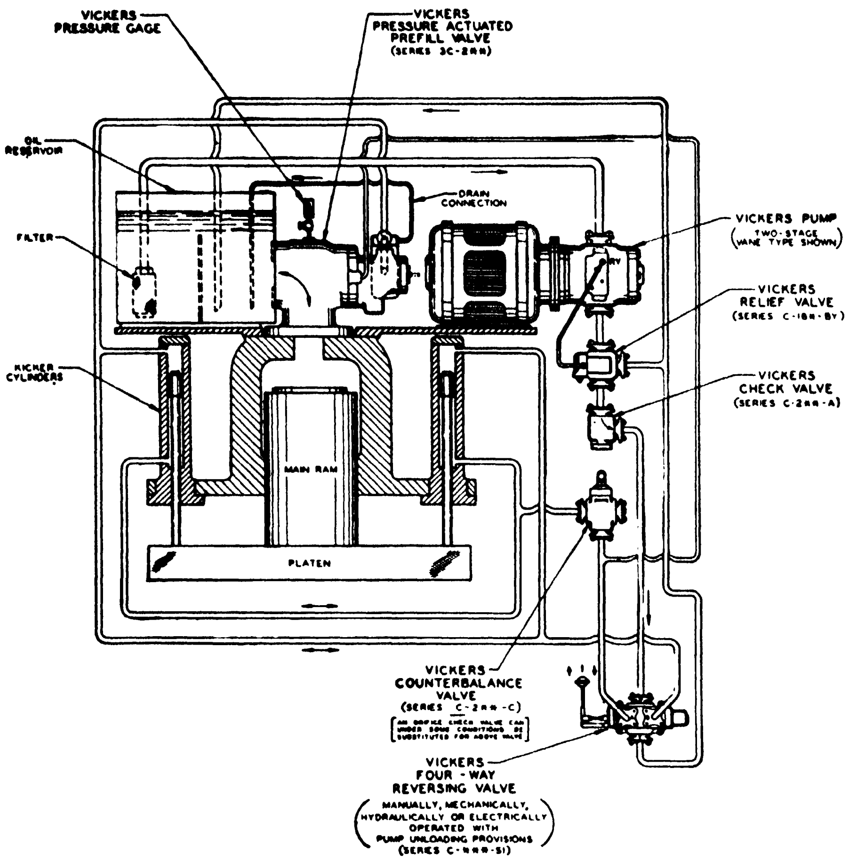


FIG. 213. Hydraulic circuit with prefill system. (*Vickers Inc., Detroit, Mich.*)

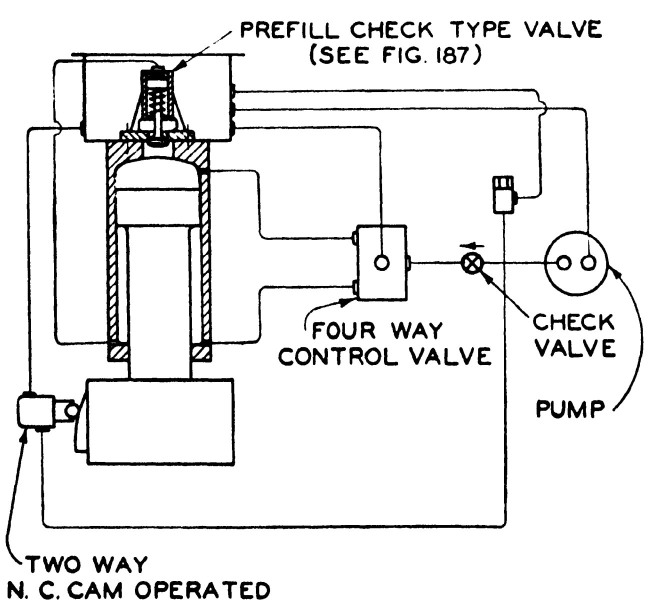


FIG. 214. Gravity rapid traverse.

weight to overcome frictional resistance, a circuit may be set up in which the kicker or booster rams are eliminated and rapid traverse is accomplished by gravity. This circuit, particularly advantageous for operation of small high-speed presses, where the platens generally have enough weight to cause gravitational descent, is shown in Fig. 214. Obviously, in this circuit the back-pressure valve must be eliminated. This means that with the operating valve in neutral or mid-position, the moving parts will not stay up. Therefore it will be found preferable to employ a spring-offset rather than a spring-centered valve and to provide separate means to keep the moving weights in suspension. To this end, a cam-operated two-way or decelerating valve is shown that is engaged by the platen at an adjustable point on its return stroke and by-passes the pump discharge to the tank. The check valve shown maintains the moving weights in suspension. This scheme has the disdavantage that there will not be a free by-pass for the pump, since the platen will come to a halt at a point where the by-pass pressure balances the moving weight. Moreover, intermediate stops between the extremes of the ram stroke cannot be made except by joggling of the valve lever and holding it in an approximately neutral position.

Both these disadvantages may be overcome by installation of a center-by-pass, blocked-cylinder-port valve (Fig. 174) instead of the valve shown. This will also eliminate the cam-operated two-way valve. Caution is indicated, however, as a blocked-cylinder-port valve should not be used on very high speed jobs (over 50 ft per min), owing to inertia, which may cause shock and hydraulic hammer. In the following, these effects and the bearing they have on hydraulic-circuit design will be discussed. Inertia effects will be felt when sudden stoppage or reversal of hydraulic rams or pistons traveling at high speed is attempted. These effects will be greatly minimized by proper design and selection of valves and correct applications to fit the requirements of the operation. For instance, if we analyze the circuits shown in Figs. 209 to 214, we find that sudden stops of the moving members by manipulation of the operating valve are avoided.

Let us assume, for instance, that the ram in Fig. 212 descends at rapid speed, and we throw the operating valve into the neutral position. At this instant the back-pressure valve is open with oil flowing from the pullback port to the valve and through the valve into the exhaust. Moving the valve to neutral does not alter this condition; all we do is to introduce a slight restriction in the oil flow to the exhaust. At the same time, as we move the valve to neutral, the flow of oil that originally passed from the operating-valve connection to the working area of the piston, is now diverted into the exhaust. The piston thus deprived of its driving power

comes gently to rest by gradual closing of the back-pressure valve, as the inertia forces spend themselves. Again in Fig. 214 it may be seen how the piston is brought to rest at the end of its return stroke by the diversion of driving fluid from the pullback to the tank rather than a sudden stoppage of fluid supply.

If, however, the valve is thrown quickly from "hard over forward" to "hard over" in reverse, the moving piston will have had no chance to decelerate, and oil pressure will be supplied to the retraction area while the piston is still moving in the forward direction. Shock and hammer are then unavoidable, if the traveling speeds are at all high. This may be avoided by care on the part of the operator, but where we must deal with careless labor or in cases where spring-offset solenoid-operated valves are used, this type of direct-actuated valving should be restricted to applications where traveling speeds do not exceed 75 to 100 ft per min. Application of blocked-cylinder-port valves (Fig. 174) is still more hazardous, as even too rapid a movement of the valve to the neutral position will cause a sudden blocking of flow from the traveling piston or ram, and use of this valve in circuits where traveling speeds exceed 50 ft per min is not recommended. All these effects may be greatly minimized by provision of preadmission or throttling grooves or V-shaped slots in the operating valves, but it must be remembered that in cases of mechanical spring- or solenoid-actuated valves, the movement may be so rapid that these expedients are unable to exercise their function.

Absolutely smooth reversals, controlled deceleration and acceleration, even at highest speeds (150 to 200 ft per min and more), may be accomplished by the use of pilot-operated valves with V slots or preadmission grooves and pilot valves to actuate them at controlled speeds through provision of adjustable chokes. A separate source of pilot pressure in these cases is desirable to maintain pilot pressure constant and ensure repetitive performance. This method of operation is indicated for fine machine tools operating at high speeds where absolutely smooth operation and absence of jerks and vibration is mandatory.

Inertia forces are also set up when rapidly moving pistons strike fixed stops on the machine or engage work at high speed. The designer should make provisions to cushion the impact or provide slowdown prior to engaging the work. For operating hydraulic pistons against stroke-limiting stops, provisions of cushioned cylinders is recommended (see Chap. VIII, Sec. 3). These should always be used when traveling speeds exceed 50 ft per min. Pistons operating at rapid-traverse speeds should be slowed down prior to engaging the work. We have seen how this may be accomplished in a two-pump system by providing a cam-operated by-pass valve for the rapid-traverse pump. A similar effect may be

produced by the introduction of a cam-operated shutoff valve in the circuit of Fig. 214. This is shown in Fig. 215, where in rapid-traverse position the ram will descend rapidly, oil discharging from the pullback connection through the open two-way valve as well as the restricted by-pass. When the cam engages the roller of the two-way valve, all oil will be forced to pass through the restriction, thus slowing down the ram to any desired degree. Similarly the operation of the circuit shown in Fig. 213 may be modified by providing shift-over from the booster

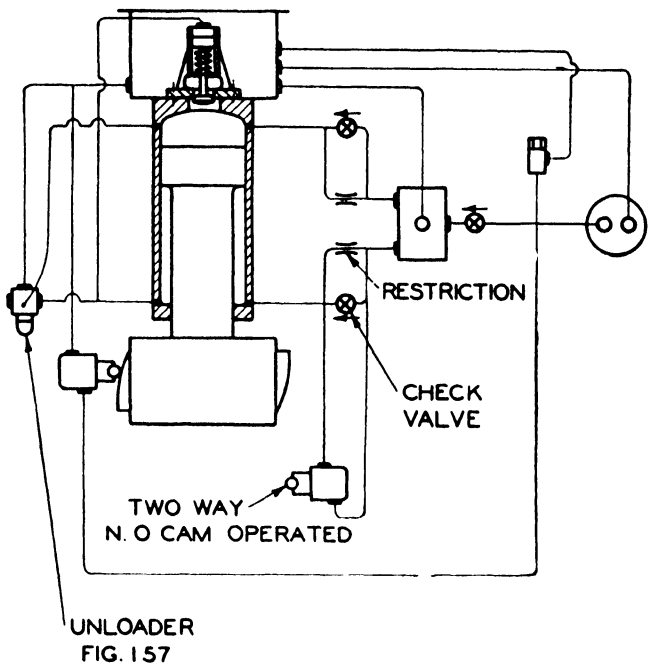


FIG. 215. Rapid-traverse circuit with slowdown and decompression feature.

cylinders to the main cylinder by a cam-actuated valve at a given position, rather than at the development of a given pressure.

In the circuit in Fig. 215, additional valving is shown, the purpose of which is to take care of the decompression of the oil in the cylinder. In Sec. 1, Chap. II, the compressibility of the operating oil has been discussed and an average value established. This characteristic is as important in the release of pressure in a hydraulic cylinder as it is in the build-up.

With directly manually actuated control valves, the reversing characteristics of the machine are left entirely to the discretion of the operator. With a properly designed valve having throttling grooves in neutral position, very smooth reversals may be obtained with reasonably experienced operators. With a valve carelessly slammed from "hard over" to "hard over," shocks and hydraulic hammer may be experienced, due both to the sudden release of the cylinder under pressure and to the sudden stoppage of flow from the pump when discharge is diverted into a stationary ram area, or, even worse, into one moving in opposite direction. In

smaller cylinders and at lower pressures, this effect is not too serious; in cases of high pressures and large volumes, it may be disastrous. Devices to eliminate this undesirable characteristic entirely in cases of manually or direct solenoid-actuated or mechanically actuated valves are shown in Fig. 215 and consist in the provision of adjustable restriction and unloading valves. If, after completion of the working stroke and with the head end of the cylinder under pressure, the operating valve is reversed, the only escape of pressure fluid from the cylinder is through an adjustable restriction, regardless how fast the operating valve may be

TABLE I. PREFERRED USES OF OPERATING—VALVE TYPES

Operating-valve type	Ram or piston speed, ft per min					
	50	75	100	125	150	200
Blocked-cylinder pump by-pass (Fig. 174), manual or solenoid	S*	D†	N‡	N	N	N
Center by-pass (Fig. 167), manual or solenoid	S	S	D	N	N	N
Center by-pass, pilot-operated, manual, automatic or solenoid pilot	S	S	S	S	S	S
Separate decompression valve	§	§	§	§	§	§

* Satisfactory.
† Doubtful.
‡ Not recommended.
§ Should be used on cylinders, exceeding 10-in. diam. if pressures are higher than 1,000 psi, or on smaller cylinders if extremely smooth reversals are desired. May not be needed on pilot-operated valve circuits.

thrown or how wide it is opened. During this decompression period the pump will circulate through an unloading valve held open by the main cylinder pressure. After decompression in the cylinder has taken place and the pressure has dropped to a safe value determined by the setting of the unloading valve, this valve will close, permitting the pump to build up pressure on the retraction area, force the surge check valve open, and return the piston.

The question naturally arises about when and where this somewhat involved scheme of decompression and by-pass valves is necessary. This question cannot be answered dogmatically, but as a general rule decompression valves should be used on cylinders 10 in. and larger, if pressures are in excess of 1,000 psi, and especially if pumps of large capacity are used. By the employment of pilot-operated valves, suitably controlled, the use of separate decompression valves may generally be avoided. The rules given in the preceding are, of necessity, very general, and exceptions may be made by an experienced designer to fit conditions

in individual cases. For the guidance of the reader who wishes to become familiar with the general limitations and preferred uses of these devices, Table I recapitulates this information in convenient form.

The circuits shown in Figs. 209 to 212 have spring-centered operating valves. When the operator releases the valve handle, the piston will stop. To return to a predetermined position, the operator must hold the valve lever on return and release it when the given position has been reached. It is desirable in many cases to have the ram return to a given position and automatically stop there. In Fig. 214 this has been accomplished by a cam-operated by-pass valve and a spring-offset operating valve. Similarly circuits in Figs. 209 to 212 may be modified by using spring-offset valves and providing for cams or control rods to pull the operating-valve stem into neutral position. While this works out satisfactorily when a foot or balancing valve is installed, or on a horizontal cylinder, caution is indicated in a circuit as shown in Fig. 214. An attempt to dispense with the separate two-way valve and actuate the control valve by means of cams or rods may result in violent oscillation at the return stop point, owing to inertia effect of the traveling piston dragging the valve past the stop position and causing reversal, followed by another return movement, etc. This vibration continues and rarely damps out.

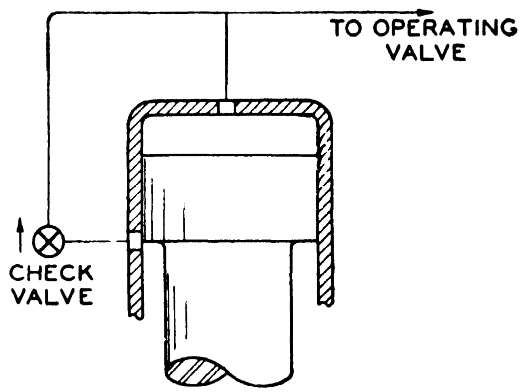
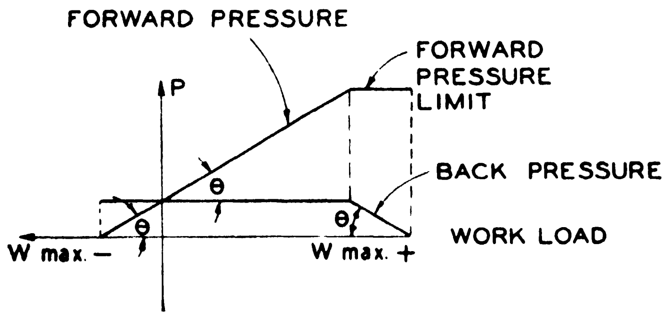


Fig. 216. Check-valve by-pass.

A simple, nonadjusting means to stop a piston at the end of its retraction stroke is provision of a check-valve by-pass as shown in Fig. 216. The piston at the end of its retraction stroke opens a by-pass, permitting the discharge to escape into the head area and from there through the operating valve into the exhaust. Upon reversal of the valve, the check valve prevents loss of operating pressure when advancing the piston.

3. The Back-pressure Circuit. When a hydraulic piston travels under pressure, the oil in the head or pressure end is compressed and elastic energy stored. As long as the resistance against which the piston operates remains constant, the piston will continue at a steady rate corresponding to the pump output or valve setting. In case of a sudden drop of resistance, as when the drill breaks through in a drill press, or when a drawn shell passes through the die at the end of a drawing operation in a hydraulic press, the compressed oil will expand suddenly, causing the piston to lunge ahead, possibly causing damage to tools or workpiece.

Similarly, in certain operations such as climb cutting with a milling machine, there is a tendency for the tool to pull the table and with it the hydraulic piston, so that control over the table feed is lost; table feed would then become a function of the cutter speed. For this type of operation, a variety of back-pressure circuits have been developed. The principle of all of them is the attempt to lock the piston between two columns of oil, and thus the term "locked circuit" was originated. However, only the circuit developed and used by the Cincinnati Milling Machine Co. is a locked circuit in the strict sense of the word. To some



$$W = (P_F - P_B)A$$

$$\tan.\ \theta = \frac{1}{A}$$

$$\tan.\ \theta = \frac{P_B}{W\,max.-}$$

$$W\,max.- = A\,P_B$$

FIG. 217. Forward and back pressure as functions of work load. Spring-loaded back-pressure valve.

extent, the back-pressure valve shown in Figs. 210 to 212 serves this function and may be used for this purpose. In Fig. 217 forward and backward pressures are shown plotted against the working load.[1] (For the sake of simplicity in this and the following diagrams, the forward and return areas are assumed to be equal.) A spring-loaded back-pressure valve of the type shown in Fig. 154 will provide a constant back pressure over the entire working range. It will provide a measure of protection against lunging, depending upon the amount of back pressure, and will support a small negative work load. Another arrangement is shown in Fig. 218, illustrating the action of the Sundstrand back-pressure circuit valve. The valve itself was shown in Fig. 196 and described in connection with the Sundstrand PWX feed unit. It ensures at all times a minimum feed pressure and can support a practically unlimited negative work load.

[1] This method of presentation has been adopted from the excellent article by A. H. Dall, Machine Hydraulics, *Machine Design*, April, 1946, 143. This method has been utilized through the courtesy of the Cincinnati Milling Machine Co., Cincinnati, Ohio.

In case of sudden drop of work resistance, a build-up of back pressure prevents jumping and lunging.

The Metered Back-pressure Circuit. The back-pressure circuits described in the preceding pages ensure maintenance of a positive pressure in the retraction area while the piston is traveling forward;

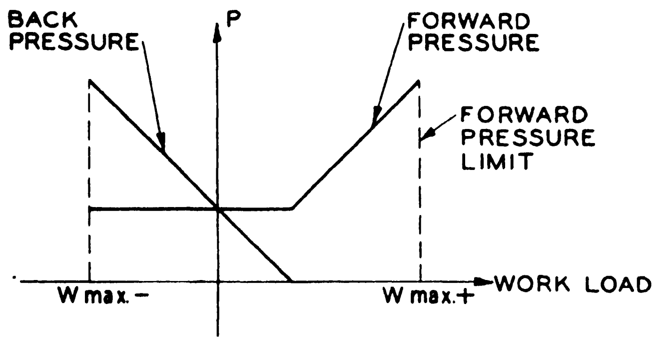


FIG. 218. Forward and back pressure in Sundstrand circuit.

they prevent lunging forward of the piston and carry negative work loads. They do not, however, serve as metering or speed-control valves; the speed of the piston is entirely governed by the amount of fluid flow supplied to the working area. Circuits that will combine metered flow with back-pressure-circuit characteristics will be described in the following.

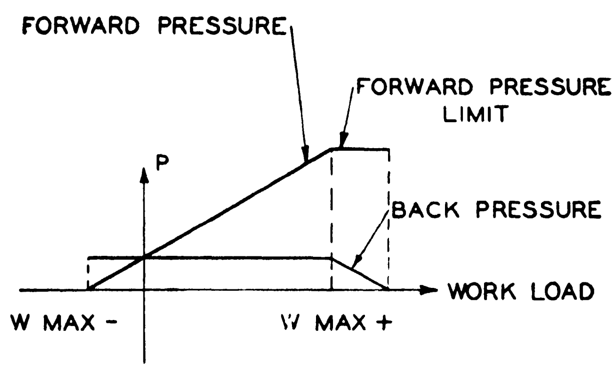


FIG. 219. Metered circuit with constant back pressure.

A constant-back-pressure circuit, similar to that shown in Fig. 217, which provides for metered outlet through a back-pressure choke, is shown in Fig. 219. The valve itself was illustrated in Fig. 190. This device will support moderate negative work loads. Pump or forward pressure varies with work load.

A metering circuit that will maintain constant forward pressure and support very large negative work loads is shown in Fig. 220. In this circuit, a metering-out type of flow control (Fig. 188) is used. This type of circuit is used in the Vickers control panel and is well adapted for drill-press, milling-machine, and other circuits. Efficiency of this circuit is

low, particularly when used with constant-delivery pumps. All the pump output not used for doing work is discharged at full working pressure, and the discharge from the back-pressure area is throttled against back-pressure resistance.

The maximum allowable system pressure occurring at maximum

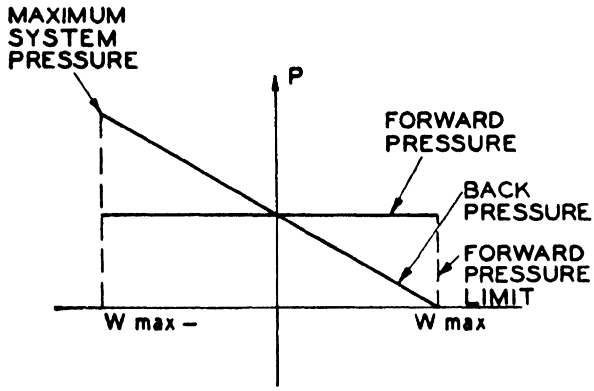


FIG. 220. Metering-out circuit.

negative work loads severely limits the maximum forward pressure, thus restricting the capacity of the machine. Most of this disadvantage has been circumvented by the differential relief valve used in the locked hydraulic circuit of the Cincinnati milling machine, as shown in Fig. 248.[1] This ingenious device is interposed in the circuit between forward- and back-pressure lines and acted upon by both pressures in a prede-

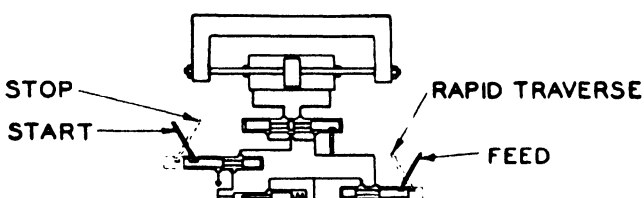


FIG. 221. Metering-out circuit with differential relief valve. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

termined ratio. Figure 221, taken from Dall's article, shows schematically a throttle-type metering-out circuit with the differential pressure relief valve interposed. The forward pressure acts on the valve plunger just as on an ordinary relief valve and escapes into the exhaust when opening the valve. The spring pressure is opposed not only by the forward pressure acting on the valve plunger, but also by the back pressure acting on an auxiliary area. The relationship between back and forward

[1] See S. Einstein and Hans Ernst, Hydraulic Feeding Mechanism for Milling Machines, *Trans. A.S.M.E.*, January–April, 1928. MSP, 50–4, 42 to 50. Also Dall, *op. cit.*

pressure depends, of course, on the ratio of the respective relief-valve areas.

Obtainable work loads may be computed as follows:

$$Ap_F = W + Ap_B \tag{1}$$

where A = area of working piston
p_F = forward pressure
p_B = back pressure

Also

$$ap_F + nap_B = ap_{max} \tag{2}$$

where a = area of forward end of relief valve
n = ratio of relief-valve areas
p_{max} = maximum allowable system pressure

From Eqs. (1) and (2) we obtain

$$\frac{W}{A} = p_F - p_B = p_{max} - np_B - p_B = p_{max} - (n + 1)p_B \tag{3}$$

Also

$$\frac{W}{A} = p_F - p_B = p_F - \frac{(p_{max} - p_F)}{n} \tag{4}$$

For any given value of n the obtainable work loads may then be computed as functions of forward, back, and maximum pressures. Maximum positive and negative work load occurs at zero back and forward pressure, respectively. Therefore

$$\left(\frac{W}{A}\right)_{+max} = p_{max} \tag{5}$$

$$\left(\frac{W}{A}\right)_{-max} = \frac{p_{max}}{n} \tag{6}$$

For $n = 1$, $(W/A)_{+max}$ becomes equal to $(W/A)_{-max} = p_{max}$, and the diagram of Fig. 222 results, which is reproduced from Dall's article.

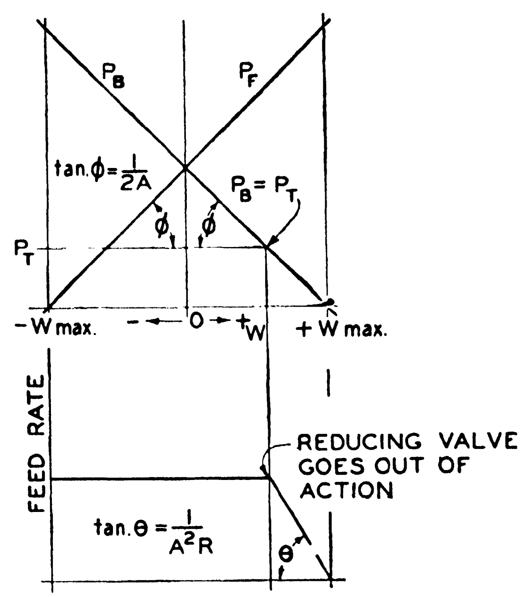


Fig. 222. Curves showing pressure relationship for circuit of Fig. 221. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

A glance at Fig. 222 will show how greatly the capacity of a hydraulic machine may be increased by the use of this ingenious device. In the ordinary meter-out circuit, the obtainable work load corresponds to only one-half the maximum system pressure. Contrasted with this, the introduction of the differential relief

valve permits operating with forward and back pressures equal to the maximum system pressure. Idling pressure is only a fraction of the available maximum forward pressure, thus the efficiency of the machine is greatly improved.

Incorporating Metering and Back-pressure Valves in Circuit. Metering and back-pressure valves may be incorporated in the circuit in a number of ways. Spring-loaded back-pressure or balancing valves are generally used to balance the moving weights of vertical machines and are installed in the pullback lines, as shown in Figs. 210 to 213. Check valves must be provided to permit return flow, either built in (Fig. 154) or separate.

Other types of back-pressure valves described in the preceding pages may be installed in a similar manner and provided with return-flow checks, if a one-directional back-pressure circuit is required. The valves may be shunted out by cam-operated two-way valves, as in the arrangement in Fig. 215, if rapid traverse of the machine is desired over certain portions of its cycle.

A two-directional back-pressure circuit is shown in Fig. 221. It may be seen that the outlet from the four-way directional valve is conducted to the meter-out valve via a two-way selector valve, selecting between feed (locked circuit) and rapid traverse. It should be mentioned in this connection that for a hookup as shown in Fig. 221, where the exhaust from the four-way control valve is subjected to system pressure, a valve of the type shown in Fig. 185 should be used to keep the outlet pressure off the stem packings. Figure 221 also shows a start-stop by-pass valve that is frequently used in machine-tool circuits in preference to a neutral by-pass four-way valve, particularly in automatic systems.

Meter-in and Bleed-off Circuits. In cases where a back-pressure circuit is not required, but adjustable constant speed irrespective of resistance is required in connection with a constant-delivery pump, meter-in or bleed-off valves are used frequently. This arrangement is shown in Fig. 223 in connection with a rotary-fluid-motor application. The circuit shows Vickers units, including their flow control (Fig. 193) with integral overload adjustment.

Meter-in circuits are notoriously inefficient as explained previously in Chap. X. They have the advantage of automatic slip compensation (in so far as the pump is concerned).

If the metering valve is put in parallel with the hydraulic motor, rather than in series, with its discharge connected to the exhaust, a bleed-off circuit results, which is far more efficient, as the pump has to supply only the pressure required to do the work at that particular moment. However, motor speed is subject to variation caused by change in pump slippage if pressure is not constant.

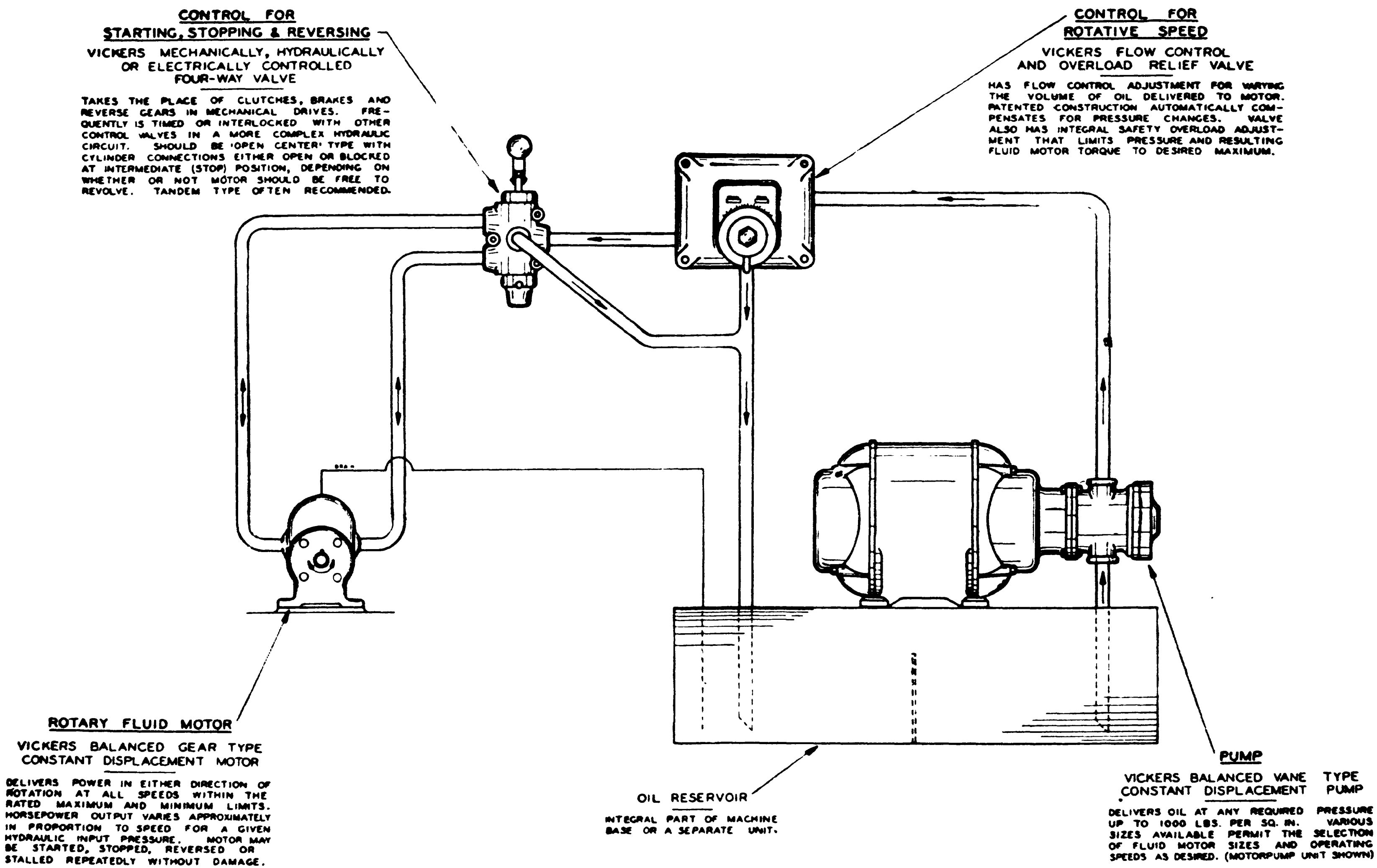
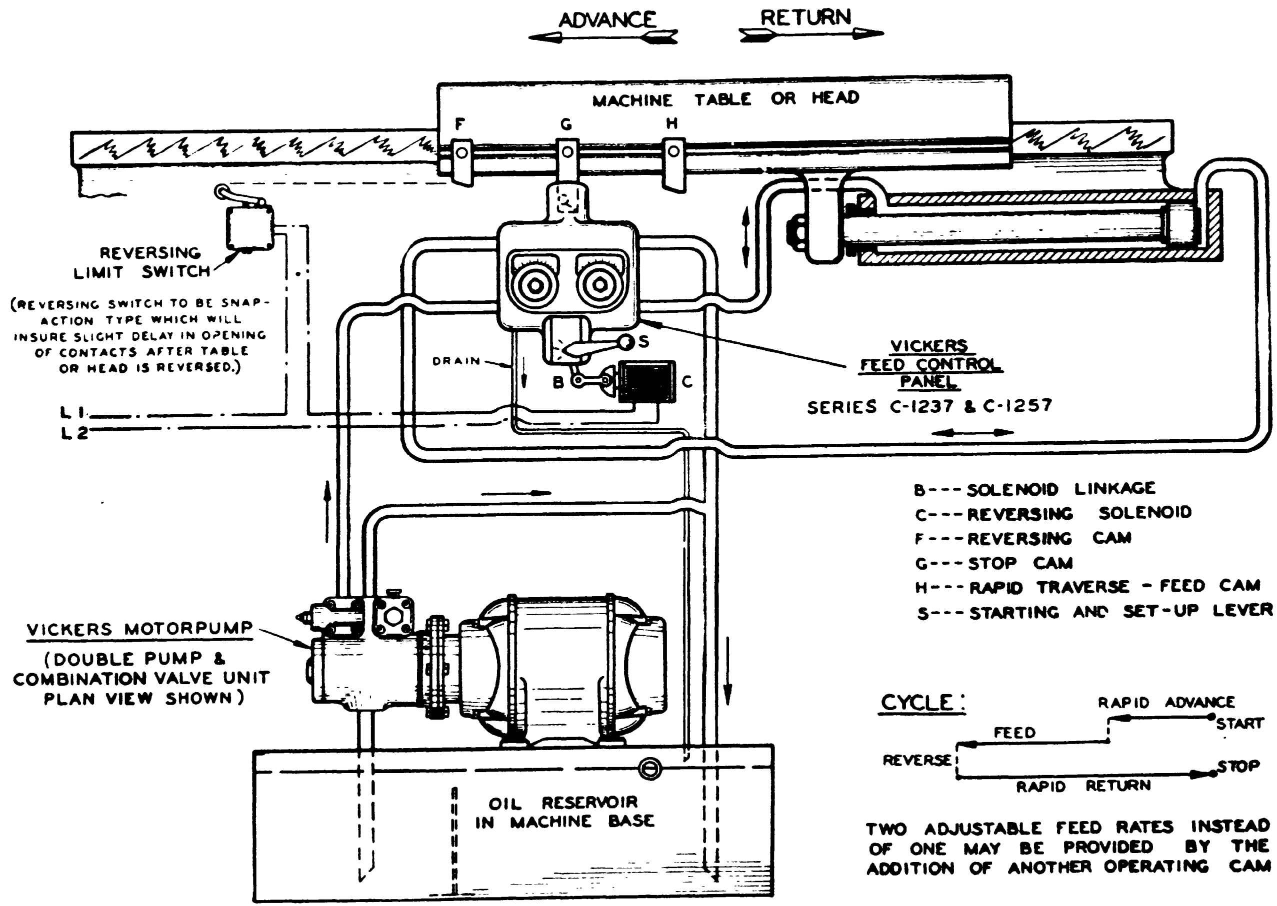


FIG. 223. Hydraulic circuit with rotary motor and metering-in valve. (*Vickers Inc., Detroit, Mich.*)

4. Automatic Hydraulic Systems. An automatic hydraulic system may be defined as one in which the sequence of operations is controlled either by the movements of the machine or the pressure generated therein, and is not subject to the will of the operator. We may distinguish between semiautomatic machines, in which each individual cycle of sequences is initiated by the operator, and full automatic machines, which continue to initiate subsequent cycles by mechanical action, either by position or pressure, and continue to repeat until stopped by the operator.

Automatic action may be produced by cams attached to the moving slides or tables that engage and actuate hydraulic valves to perform the sequence of operations. It is axiomatic that a reversing movement may not be produced by the mechanical actuation of a direct-actuated valve, so that a force other than that produced by the movement of the slide propelled by the hydraulic piston must be available to do this. Electric solenoids, pilot-actuated cylinders, or mechanical energy stored in a "load-and-firing" mechanism may be used for this purpose. Standard machine-tool panels offered by a number of manufacturers fall into this category.

Figure 224 shows a control circuit employing the Vickers feed-cycle control panel illustrated in Fig. 195. The functioning of the panel itself was fully described in Sec. 5, Chap. X, and we shall confine ourselves to a description of the circuit. The illustration shows a high- and low-pressure pump combination with built-in by-pass and relief valve (see Fig. 67), mounted upon the end bell of the driving motor. The suction of both pumps is common. Oil is discharged to the intake of the control panel, where it is directed, by manipulation of a suitable multiple-head directional valve, to the forward or retraction area of the table cylinder. Exhaust from the corresponding opposite end is routed to the tank, either directly, when on retraction stroke, or optionally through metering-out valves, or directly when on feed or rapid-traverse stroke. The main directional valve is connected to the hand lever shown in front and may be actuated by this lever for setting-up purpose or to start the cycle. The valve may also be actuated by suitably arranged cams on the table to permit the machine to proceed with an automatic cycle, as will be described in the following. By mechanically operating the lever *S* on the panel, the control valve is raised to its uppermost position, causing rapid-traverse movement of the table. As soon as dog *H* depresses the valve plunger to the next lower position, the outlet from the differential end of the cylinder is routed through the metering valve in the panel, and feed stroke begins. If desired, a second dog may be provided to produce a second feed rate, utilizing the second metering valve on the



ABOVE DIAGRAM INDICATES A "SEPARATE SOLENOID" INSTALLATION. VICKERS PANELS OF SIMILAR DESIGN BUT HAVING INTEGRAL SOLENOID CONSTRUCTION ARE ALSO AVAILABLE.

FIG. 224. Hydraulic circuit with feed-cycle control panel. (*Vickers Inc., Detroit, Mich.*)

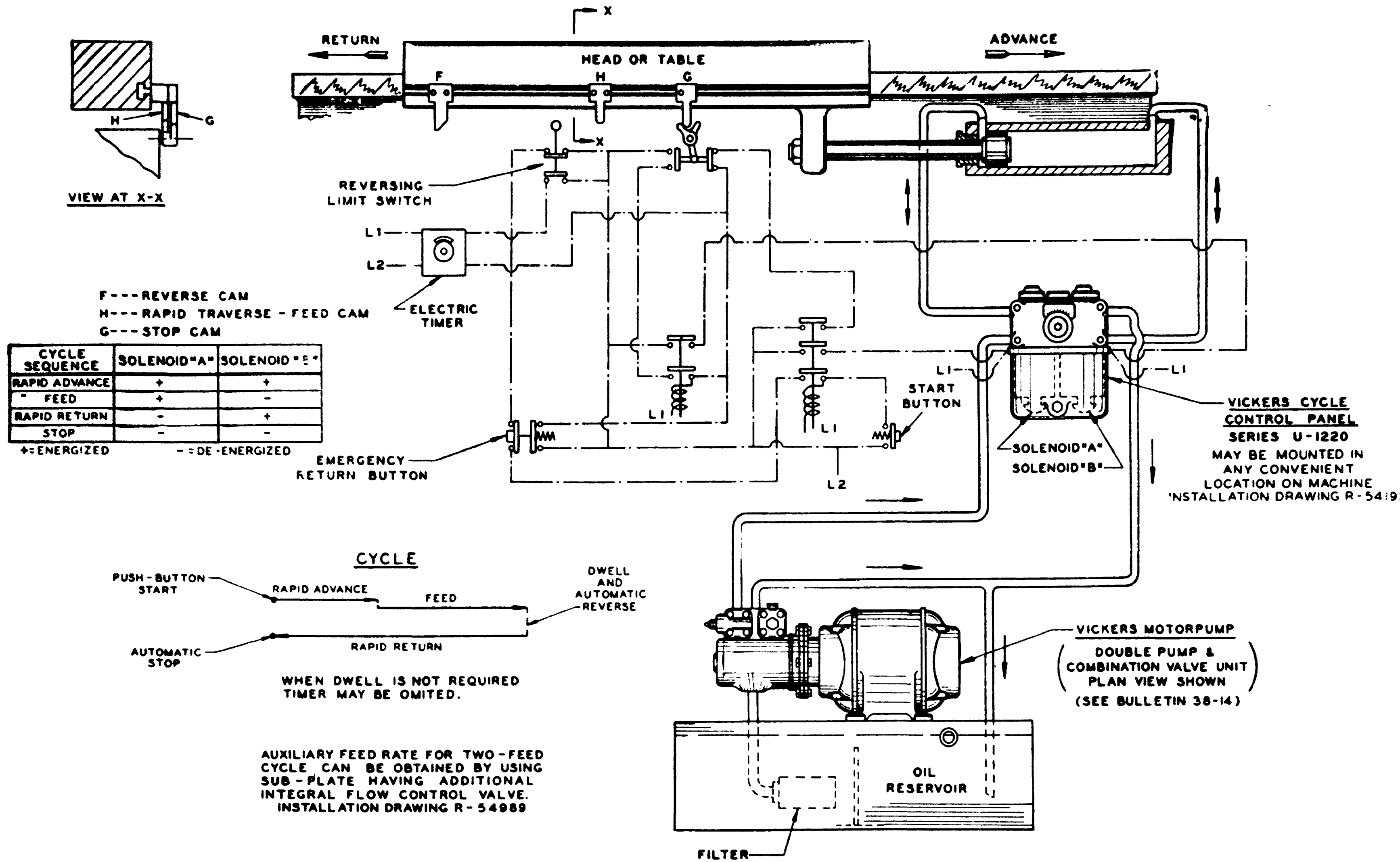


FIG. 225. Hydraulic and electric circuit of Vickers control panel. (*Vickers Inc., Detroit, Mich.*)

panel. Finally, reverse dog *F* strikes limit switch, energizing solenoid *C*, which returns the valve plunger to its lowest position, and the table will reverse. It will continue on the retraction stroke until a hook-type cam *G* engages the valve plunger, returning it to the neutral or pump-by-pass position, and the table will come to rest, ready for the next cycle.

A similar circuit may be set up with a forward-stroke-starting solenoid in addition to the reversing solenoid shown in Fig. 224. Then the cycle may be initiated by the operator, depressing a cycle-start button to

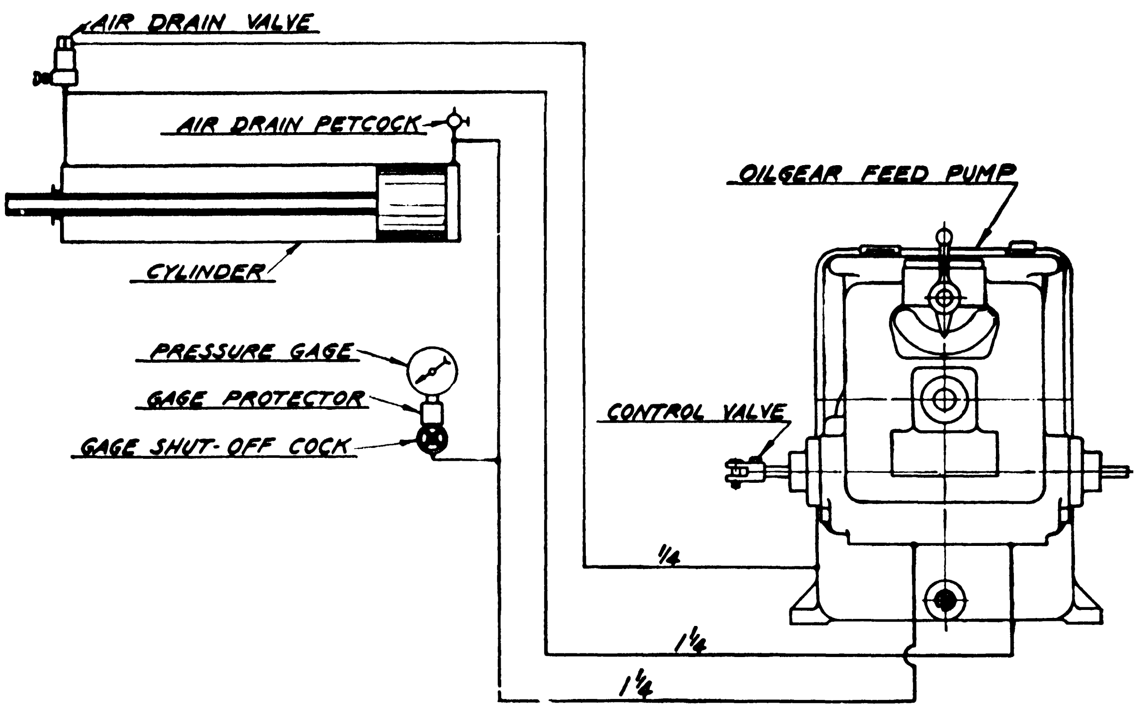


FIG. 226. Oilgear fluid-power feed circuit. (*The Oilgear Co., Milwaukee, Wis.*)

elevate the valve plunger to its uppermost position; thereafter the cycle proceeds in the same manner as explained above. Mention should be made of the fact that both solenoids are only momentarily energized and will deenergize when the respective switches are released; no electrical interlock is used. The valve plunger is held in its different positions by spring-loaded detents, which prevent inadvertent displacement. A great number of variants is possible with a panel of this kind by different arrangement of valve plunger, dogs, and electrical devices. Full automatic reciprocating cycles may be obtained by dogs on the table engaging limit switches at both ends of the table travel; feeds may be had in both directions, and other modifications to suit any particular circuit setup may be made.

Vickers panels may also be supplied with reversing movement of valve plunger, obtained by a pilot cylinder, actuated by a cam-controlled pilot

valve. In all other respects the circuit is similar to that shown in Fig. 224.

Automatic operation may be produced by completely electrical actuating means. Vickers supplies feed panels for this purpose with electrical-solenoid-actuated valves. Figure 225 illustrates a circuit with an electrically controlled directional-control and metering panel.

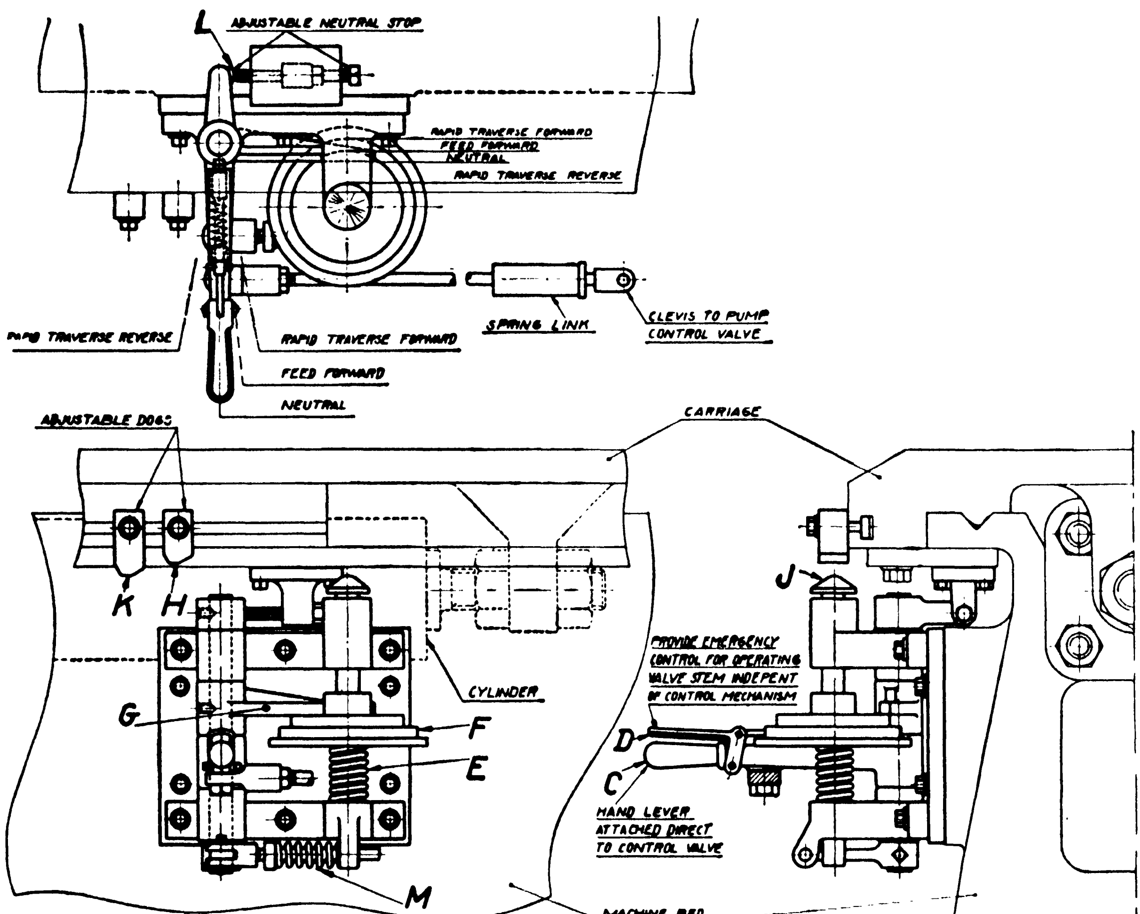


FIG. 227. Suggested control mechanism for fluid power feed. (*The Oilgear Co., Milwaukee, Wis.*)

In contrast with the mechanically actuated panel, the full electrically controlled circuit provides for electrical interlocks to maintain the solenoids in their energized state after the actuating switches have been released.

Oilgear fluid power feeds, described in the preceding chapter, may be incorporated in an automatic machine-tool feed circuit, requiring merely two pipe lines to connect the unit to the table-operating cylinder. Figure 226 shows the extreme simplicity of this arrangement. Suitable dogs or cams must be provided on the table of the machine tool to actuate the control stem in proper sequence. Figure 227 shows an arrangement

suggested by the manufacturer, which will be described in the following to acquaint the reader with the design of load-and-fire mechanism required to obtain automatic reversal of a hydraulic machine by mechanical means.

With the semiautomatic dog control mechanism in Fig. 227, operator starts the cycle by moving lever C to the rapid-traverse-forward position. Spring E lifts cam F and pin J, allowing lever G to rest against the rapid-traverse-forward surface on cam F. Operator then releases control lever. Moving head travels forward at rapid-traverse speed until adjustable dog H strikes pin J, automatically depressing cam F and allowing lever G to trip to the feed-forward position. Head feeds forward until adjustable dog K strikes pin J, again depressing cam F and allowing lever G to trip to the rapid-traverse-reverse position. As head returns, the adjustable stop L moves control lever C and lever G to the neutral position, automatically stopping the cycle. Operator can stop or reverse motion of moving heads during any portion of the cycle by depressing latch D and moving lever C to the proper position. Spring M loads lever G.

Oilgear fluid power feeds may be actuated by pilot cylinders and cam-actuated pilot valves, as well as electrically by suitable solenoids in a manner similar to that described in connection with the Vickers unit (see Fig. 225).

Automatic circuits are not confined to the use of standardized control panels, but may be constructed with any desired number of individual control devices to obtain any required sequence of operation. Again, valves may be used, directly actuated by dogs or cams on moving members of the machine, employing load-and-fire mechanism for automatic reversal by position or pressure limit built up in the system. More frequently used at present are electrical, hydraulic pilot, and electrical hydraulic pilot devices; these will be dealt with in the following.

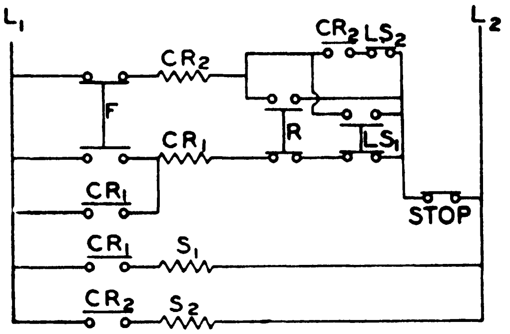
Any hydraulic circuit may be made automatic by providing solenoid-operated valves in place of manual valves and providing an electrical system to actuate these solenoid valves in their desired sequence. The circuit in Fig. 209, for instance, may be made automatic by providing a double-solenoid spring-centered four-way control valve instead of the manual control valve shown. Figure 228 shows a suggested electrical circuit to operate the valve in the following sequence:

1. Depress forward push button, energizing contactor CR_1. This contactor closes two contacts, one of them locking in around the push button, the other energizing solenoid S_1.
2. The piston travels on working stroke with solenoid S_1 energized.
3. At the end of the working stroke, a cam on the table engages limit

switch LS_1, which deenergizes contactor CR_1, contacts CR_1 drop out, and solenoid S_1 is released.

4. At the same time contactor CR_2 is energized by the double-throw limit switch; this closes two contacts CR_2, locking in the contactor and energizing return-stroke solenoid S_2.

5. The piston returns until a cam on the table engages a limit switch LS_2, which releases contactor and relays CR_2, whereby the solenoid S_2 is deenergized, and the operating valve centers itself by means of the centering springs, and the table comes to rest.



LEGEND
CR1 FORWARD CONTACTOR
CR2 REVERSE CONTACTOR
S1 FORWARD SOLENOID
S2 REVERSE SOLENOID
F FORWARD PUSH BUTTON
R REVERSE PUSH BUTTON
LS1 REVERSING LIMIT SWITCH
LS RETURN STOP LIMIT SWITCH

FIG. 228. Electric control circuit for automatic operation of hydraulic system.

A separate stop button is provided whereby the table may be stopped at any point of its forward or reverse travel. LS_1 may be actuated by a pressure-responsive plunger from the hydraulic system rather than by a table dog, thus providing pressure-controlled, instead of position-controlled, reversal. An emergency reverse push button is shown that permits reversal of the table at any point of its forward travel. The system may be made fully automatic by introduction of a suitable limit switch in place of or parallel with the forward button. Selector switches may be provided to select any desired option of operation and/or mode of reversal.

Any hydraulic circuit may be electrified by use of suitable solenoid-operated control valves and an electrical circuit diagram set up to produce any desired sequence of operation. Time delays may be easily handled by use of electrical timing devices, which are available in a great variety of designs and styles. Thus the motions of a hydraulically actuated machine may be completely electrically controlled; the electric controls, circuit, and wiring could be likened to the brain and nervous system in the body and the hydraulic system to the sinews and muscles.

5. Pilot-controlled Hydraulic Circuits. Control of the main valves in a hydraulic circuit by pilot valves offers many advantages. As has been pointed out, pilot-valve operation ensures control of inertia forces at even the highest speeds obtainable by hydraulic means. This is accomplished by a combination of preadmission slots or throttling grooves in the main

valve pistons and a controlled shifting speed by control of pilot fluid admitted to the valve-shifting pistons. A second advantage, particularly important in the operation of large valves, is reduction of effort to operate the main valves. A small, easily manipulated valve is actuated by the operator with the least possible amount of effort and fatigue. The pilot valve may be mounted in a convenient and accessible place for the operator and the main valve located where it is most advantageous for operation in the circuit. Not the least of the advantages is the possibility of obtaining full and semiautomatic operation of the hydraulic machine, together with suitable interlocks, without the aid of auxiliary apparatus, such as load-and-fire mechanism and electrical means, although the latter are sometimes used in conjunction with pilot valves for operational advantages, such as employment of push-button control, electric timers, etc. Pilot pressure may be obtained directly from the main source of supply by passing the supply through a spring-loaded check or equivalent sequence valve. This will be feasible only if the main supply is not so large in volume as to cause excessive loss of power and heating. In the latter case, an independent source of pilot pressure is recommended. The same is true if very accurate duplication of pilot actuation is required, depending on the maintenance of a constant pilot pressure. Often the output of a separate pilot pump may be used to operate coolers, filters, or auxiliary hydraulic devices, as long as these auxiliary functions do not interfere with its main purpose, the supply and maintenance of pilot fluid and pressure for the operation of the main system control valves.

Any of the circuits shown in Figs. 209 to 215 may be operated by pilot-actuated valves. If any of these circuits are equipped with single-pilot-piston actuated valves, controlled by standard four-way pilot valves, such as Vickers (Figs. 177 and 178), then only two positions of the main valve are possible, resulting in forward or reverse movement of the main valve. Pilot-valve movement may be by hand lever or dogs on the moving table, resulting in full automatic reciprocation. In either case, an auxiliary stop or by-pass valve must be provided to stop the table in any position, as no means are provided to put the main valve in neutral position and hold it there. Auxiliary by-pass valves may be eliminated, if the main valve is made of the self-centering type (Fig. 181) and operated by a center-pressure pilot valve (Fig. 168, bottom row), or by making it spring-centered and using a blocked-center pilot valve with both cylinder connections exhausted in neutral (Fig. 168, second from bottom). In both latter cases, a standard Vickers rotary-type pilot valve cannot be used, as all connections are blocked at the crossover point of this valve.

A circuit, basically identical with that of Fig. 209 but pilot-valve-controlled, is shown in Fig. 229. This circuit provides for reversal of ram

movement by hand-lever-actuated pilot valve or dogs on the table actuating the pilot valve at both ends of the stroke. No stop valve is shown, but one may easily be added by provision of by-pass valve in the pump pressure line, or of pilot vent valve for the Vickers relief valve (see Fig. 153).

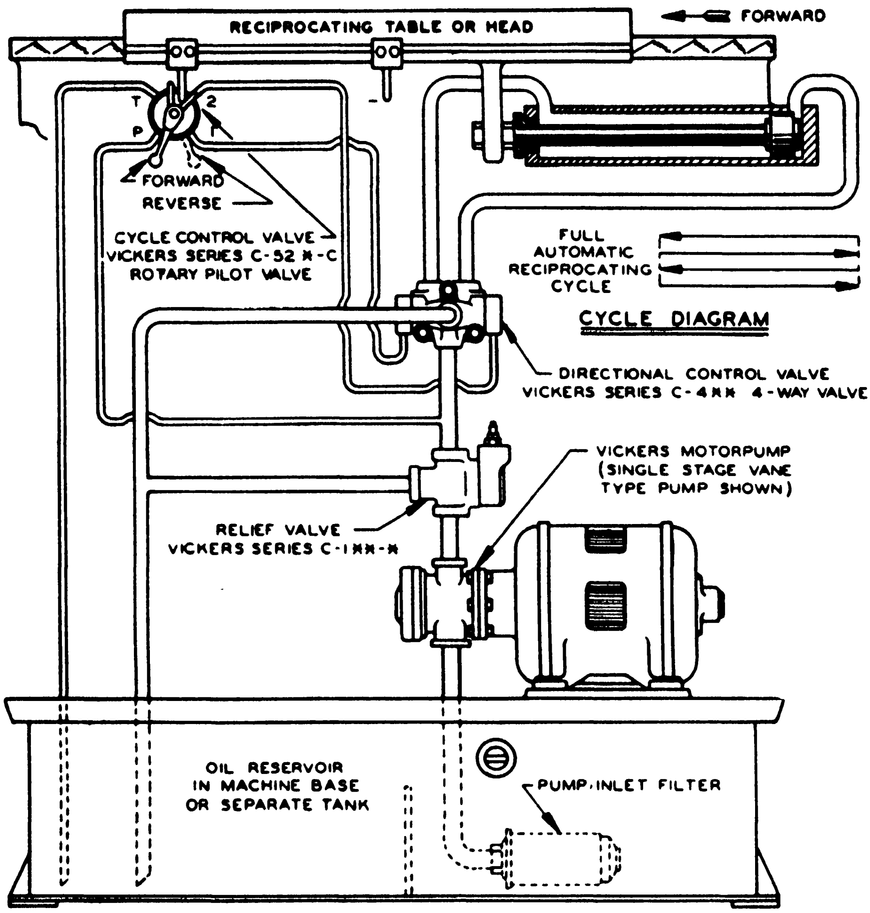


FIG. 229. Full automatic reciprocating circuit with pilot valve. (*Vickers Inc., Detroit, Mich.*)

A semiautomatic circuit, providing manual start, automatic reverse, and stop by utilizing a double-deck Vickers rotary pilot valve and a cam-actuated piston-type pilot valve, is shown in Fig. 230. In the position shown, the table is at rest with the plunger-type pilot-valve stem depressed and the rotary double-deck pilot valve in "return" position. In this position the connections 2 on the pilot valve are connected to T and connection 1 to P. Connection 1 and T are connected on the plunger valve. Pilot oil passes from the center connection of the relief valve to connection P, to connection 1 to the operating-valve flange. The internal arrangement of the pilot-operated valve is similar to Fig. 185, so that movement of the plunger to the right will admit pressure to the pullback area of the table cylinder. A line leads from the vent connection

of the Vickers relief valve to connection P on the second deck of the rotary pilot valve, thence to connection 1 to 1 on the plunger pilot valve, thence to T and the oil tank. Thus the relief valve is vented to the atmosphere, and the table is at rest with the pump idling at practically zero pressure. By placing the rotary pilot valve in "advance" position, the vent con-

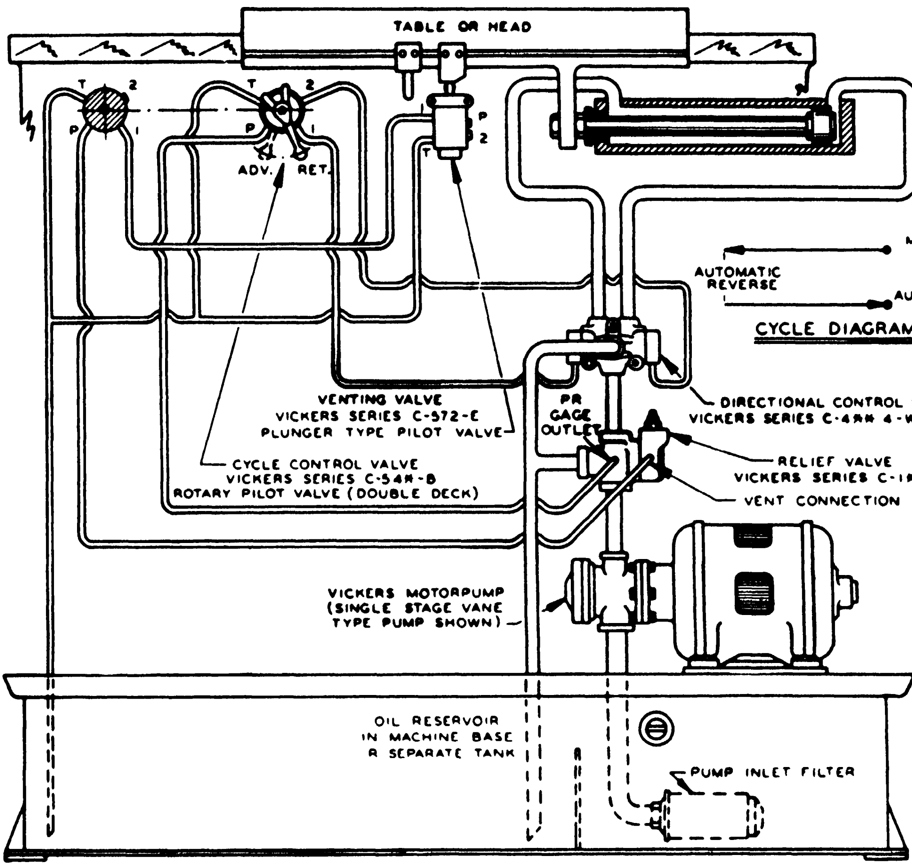


FIG. 230. Semiautomatic reciprocating circuit with rotary and piston-type pilot valves. (*Vickers Inc., Detroit, Mich.*)

nection is blocked; simultaneously pressure is applied to the opposite end of the operating valve, reversing the valve spool and with it the table travel, and the table will advance until the reversing dog engages the pilot-valve lever, again reversing the pilot valve and with it main valve and table stroke. Simultaneously the vent connection on the second deck is opened, but this will have no effect, as the stop cam in the meantime has released the roller on the plunger pilot valve, which caused this valve to block the vent connection. At the end of the retraction stroke, the stop cam will again engage the pilot-valve plunger and open the vent connection to the tank, thus bringing the unit to rest.

By a suitable combination of pilot-operated valves and pilot valves almost any desired hydraulic cycle function may be obtained. Standard

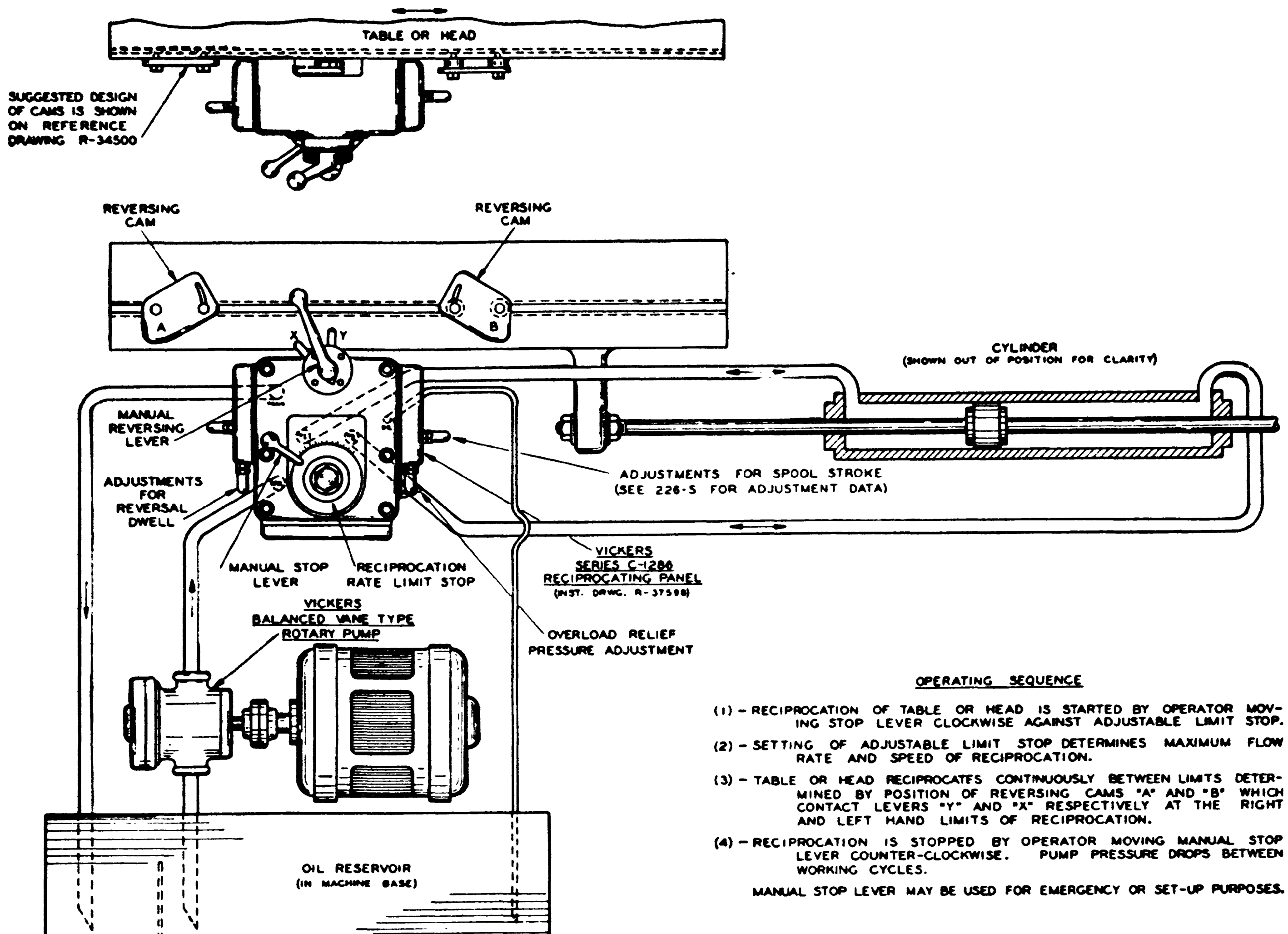


FIG. 231. Application of Vickers reciprocating control panel. (*Vickers Inc., Detroit, Mich.*)

control panels are available that permit automatic actuation by built-in pilot-actuated valves and pilot valves. Figure 231 shows an automatic circuit showing application of a Vickers reciprocating control panel. This panel contains a pilot-operated valve with built-in choke checks and spool stops to control rate of acceleration and deceleration, a rotary Vickers pilot valve, a metering valve to adjust flow rate of pump and with it reciprocating rate of table, and a manual vent valve to stop reciprocation of unit.

Extensive use of pilot valves is made in Sundstrand hydraulic machine-tool circuits. A description of the PWX and PW pumps was given in

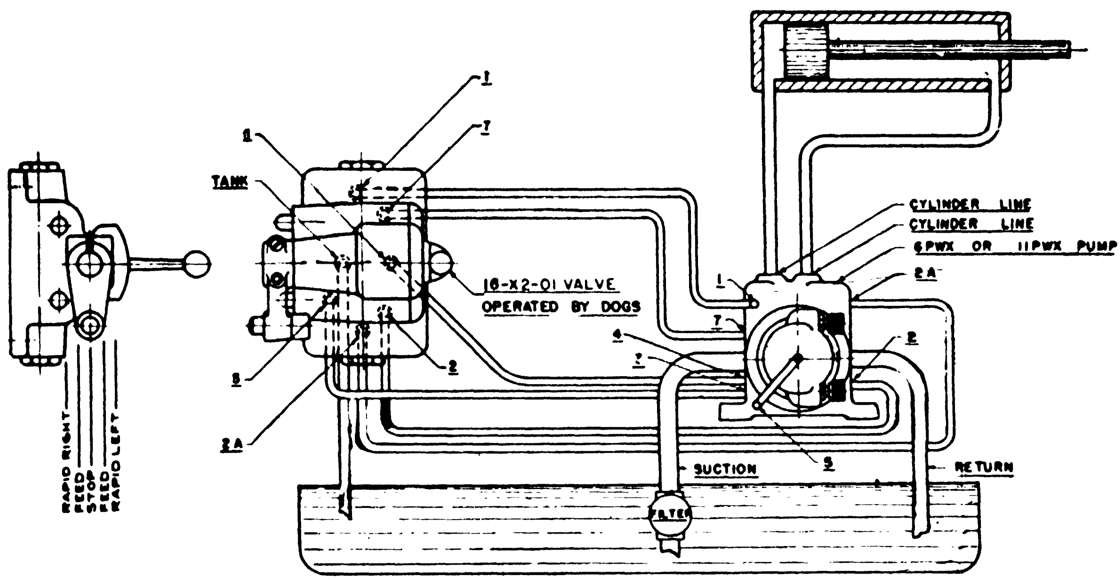


FIG. 232. Sundstrand feed circuit with two feed rates in two directions. (*Sundstrand Machine Tool Co., Rockford, Ill.*)

Chap. X. By suitably arranged pilot valves almost any combination of feed and rapid-traverse movements may be had. Figure 232 shows a circuit employing two feeds and rapid traverse in both directions. The 16X pilot valve, shown in Fig. 197, is used to control the movements of the table in a predetermined sequence. Valves integral with the PWX unit are actuated in a manner described in Chap. X, and the reference numbers coincide with those used in Fig. 232.

Instead of using a 16X pilot valve, solenoid-actuated multiple units may be employed, giving identical results with somewhat different means. This arrangement is shown in Fig. 233.

6. Reversing Pump Circuits. While the majority of present-day hydraulic machines employ directional-control-valve circuits, there is a fertile field of applications for reversing or two-way pump circuits employing variable-delivery, reversible-discharge, axial or radial pumping units.

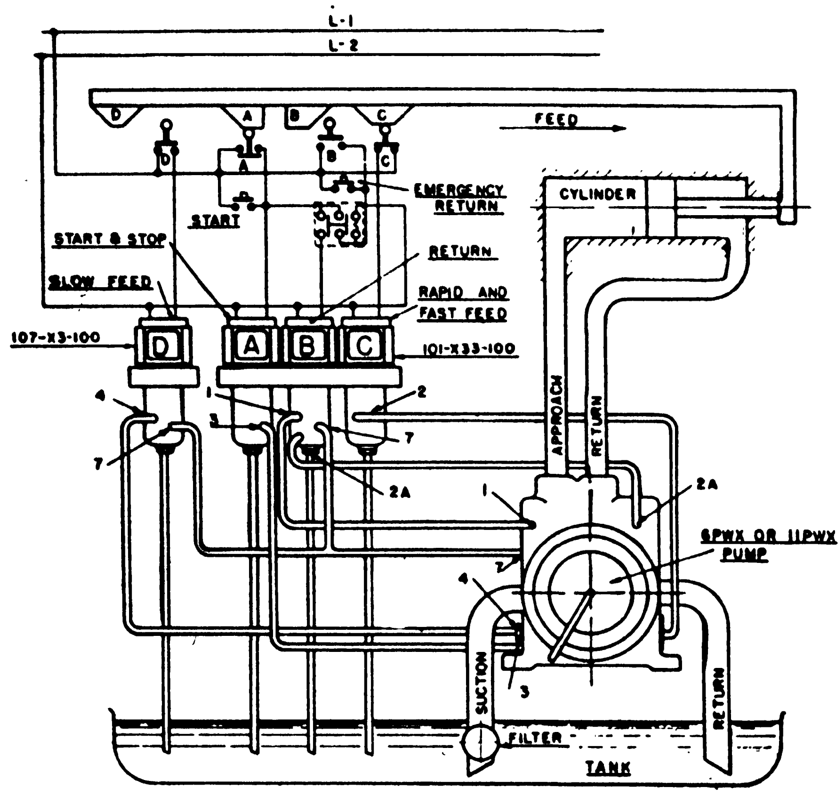


FIG. 233. Sundstrand solenoid-pilot-valve-controlled feed circuit. (*Sundstrand Machine Tool Co., Rockford, Ill.*)

In these circuits, reversal of the hydraulic motor, be it of the rotary or linear type, is accomplished by reversal of the pump-stroke crosshead rather than by valves. This type of operation has outstanding advantages, particularly for heavy-duty high-pressure applications, where smooth reversals, free from hydraulic shock and inertia effect, are desired. High-speed oil hydraulic presses and heavy machine tools fall into this category. Another important advantage is the possibility of accurately controlling motor speeds without throttling valves, and this advantage has led to the almost exclusive employment of this mode of operation for ship steering gears, windlasses, and other closely controllable heavy-duty devices.

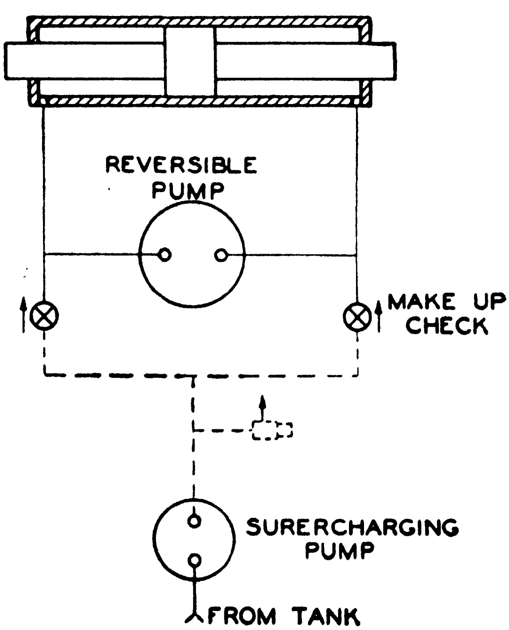


FIG. 234. Two-way pump circuit.

A two-way or reversing pump circuit may be made up in a simple manner by connecting the two lines leading from the reversible-discharge

pump to the two connections on a hydraulic motor of the rotary or linear type. If this is done, the circuit shown in Fig. 234 will result. The hydraulic motor shown is a cylinder with equal displacement in both directions; a rotary motor may be connected in the same manner. Make-up checks as shown are necessary to care for leakage losses in the pump so that the circuit may be maintained full of oil. A supercharging pump, as indicated, is desirable to keep a slight head of pressure on the system at all times and prevent air infiltration. A unit of this kind is illustrated in Fig. 116. A circuit as shown in Fig. 234 is commonly used in ship steering gears, windlasses, etc., in connection with rotary motors or rudder-actuating cylinders. Control of pump is by servo motor, generally hooked up with the rudder through follow-up linkage, so that rudder displacement will be proportional to servo-valve displacement. The arrangement of Fig. 234 works well with cylinders of equal displacement. If the displacements of both ends are unequal, as is the case in most applications, a modified circuit must be used. To this end, a shuttle or differential valve must be installed between the two pipe connections. Figure 235 shows a circuit with a double check-type differential valve. The functioning of this valve depends upon the flow of oil through it as the pump discharges in the corresponding direction. This flow closes the valve by hydrodynamic action, whereby connection is made at the opposite end between cylinder and tank, and either make-up oil may flow into the pump from the tank, or excess oil may discharge from the cylinder into the tank, depending upon whether the pressure flow is into the large or small area of the cylinder. A glance at Fig. 235 will show that the employment of this valve will automatically provide an excellent means for decompression of the oil. With one end of the cylinder under pressure, the differential valve on that end will be held closed and, upon reversal of the pump, will not move until all this pressure has been relieved by pumping the oil out of the pressure and discharging it through the open end of the differential valve into the oil tank.

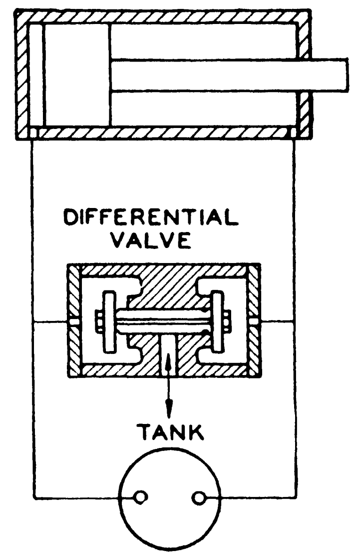


FIG. 235. Two-way pump circuit with differential valve.

A differential valve of the type described has the disadvantage that its action is entirely hydrodynamic, that is, its operation depends upon being dragged into position by the stream of oil flowing against the valve disk. For this reason the valve is not suitable for very large differentials in area. To explain this, if the pump discharges into the small one of two

disproportionally different areas, the volume escaping from the large one is so great that a very large differential valve would be required to accommodate it. Such a valve would have too low an oil velocity to be shifted by the pump discharge. Differentials of about 3:1 are about all that a valve of this kind may accommodate. At best, the forces available

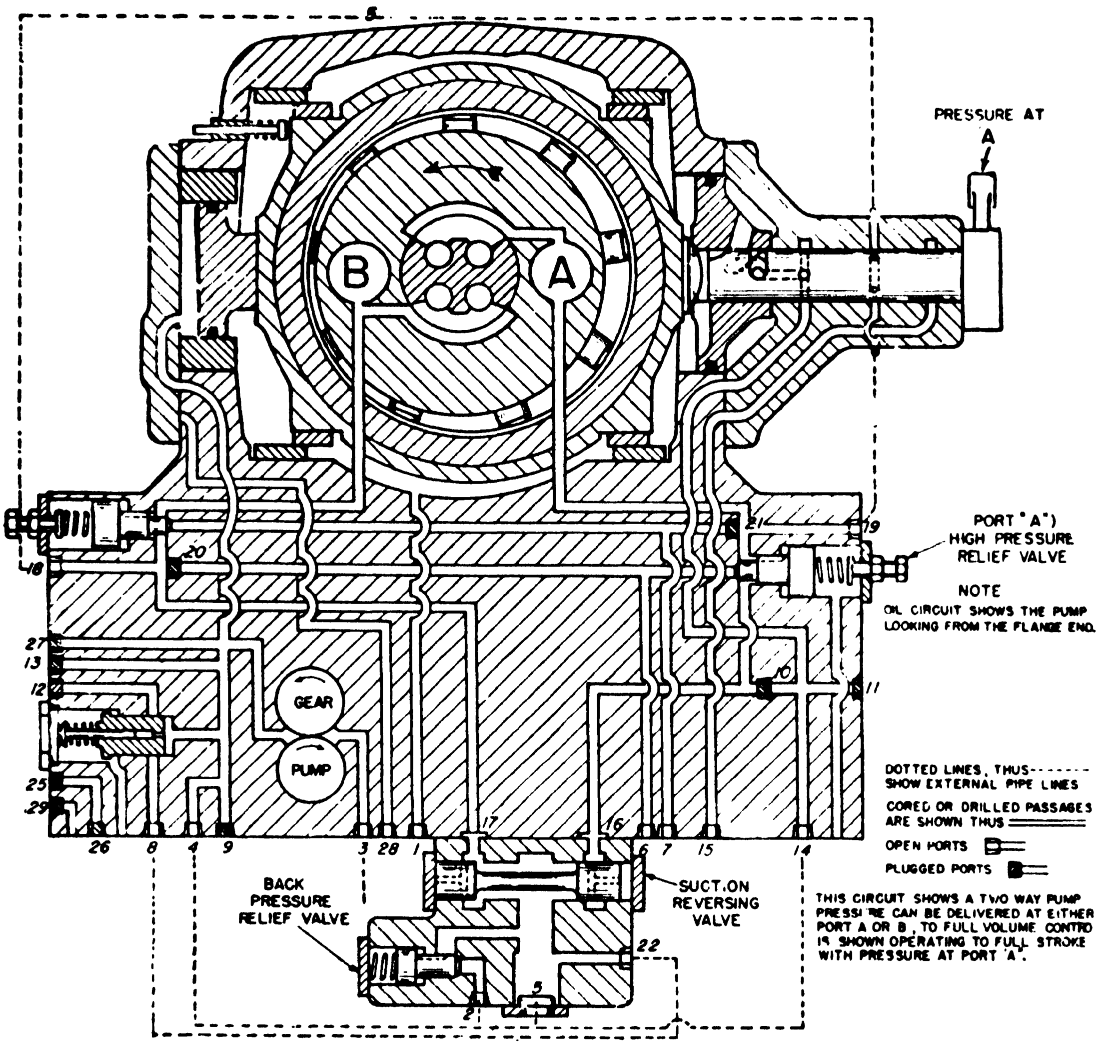


FIG. 236. Oil-circuit diagram of an Oilgear two-way variable-delivery pump with built-in gear pump, two-way automatic suction-reversing valve, relief valves, and internal passages. (*The Oilgear Co., Milwaukee, Wis.*)

to shift the valve are infinitesimal, and the slightest sticking or binding results in an inoperative machine. The automatic reversing valve developed by the Oilgear Co. avoids this objection. Combined with the action of the exhaust-pressure relief valve, a positive shifting force is available, governed by the pressure setting of this relief valve.

Figure 236 shows the oil circuit of an Oilgear two-way variable-delivery pump with built-in gear pump and two way automatic suction-reversing valve. On a simple pump and cylinder application, port *A*

is usually connected to the ram end of cylinder and port *B* to the piston end of cylinder. Ports *A* and *B* are also connected to ports 16 and 17, respectively, by cored passages. In the illustration, oil is delivered out port *A* to ram end of cylinder, as port 16 is blocked when back-pressure oil automatically shifts suction-reversing valve to the left. Oil returning to port *B* from the piston end of cylinder enters the pump intake. Excess oil flows down the cored passage through port 17, past the suction-reversing plunger and the back-pressure relief valve, and through port 2 to the reservoir. Excess oil from the gear pump flows past the gear pump and relief valve, through ports 8 and 22, past the back-pressure relief valve, and through port 2 to the reservoir.

When control lever is moved to the left of neutral position, oil from the gear pump flows through the control shaft and spiral groove into the large control-piston chamber to move slide block to the left. This causes oil to be delivered out port *B* to piston end of cylinder, as port 17 is blocked when back-pressure oil automatically shifts suction-reversing valve to the right. Oil returning to port *A* from the ram end of cylinder flows into pump intake. Excess oil from the gear pump flows in port 22, past the suction-reversing plunger, and in port 16 to port *A* to the intake. If additional intake oil is required, the radial piston pump sucks it directly from the reservoir through check valve at port 5 to port *A*.

Movement of the suction-reversing valve is by a positive hydrostatic force. Valve will not move to the right until the pressure in ports *A* and 16 has dropped below the back-pressure-relief-valve setting. It will not move to the left until the pressure in ports *B* and 17 has dropped below the back-pressure-relief-valve setting. This action, combined with decelerated and accelerated reversal of the oil flow as the slide block moves to the opposite side of center, ensures smooth decompression at a controlled pumping rate. No pressure can be applied to the opposite end of the cylinder, until all the compression has been removed by the action of the pump.

The speed of decompression in this circuit, as well as in that described in Fig. 235, will depend upon the relative size of pump and cylinder. Means may be incorporated in the oil circuit to speed up decompression by releasing oil through specially arranged control ports or pilot-actuated decompression valves. On hydraulic production presses, where the users insist on high speed of operation, these methods are frequently used.

Another make of decompression valve, combined with an automatic pump by-pass, is illustrated in Fig. 237 and fully described in Sec. 2, Chap. XII.

Any one of the circuits shown in Figs. 234 to 236 will work with a vertical as well as with a horizontal cylinder. If the ram is permitted

to drop by gravity, the speed of descent is limited by the rate at which the oil is removed by the pump. Therefore, reversing pump circuits are inherently locked circuits, but only in the sense that there is an upper limit to the speed at which a piston under pressure may lunge ahead in

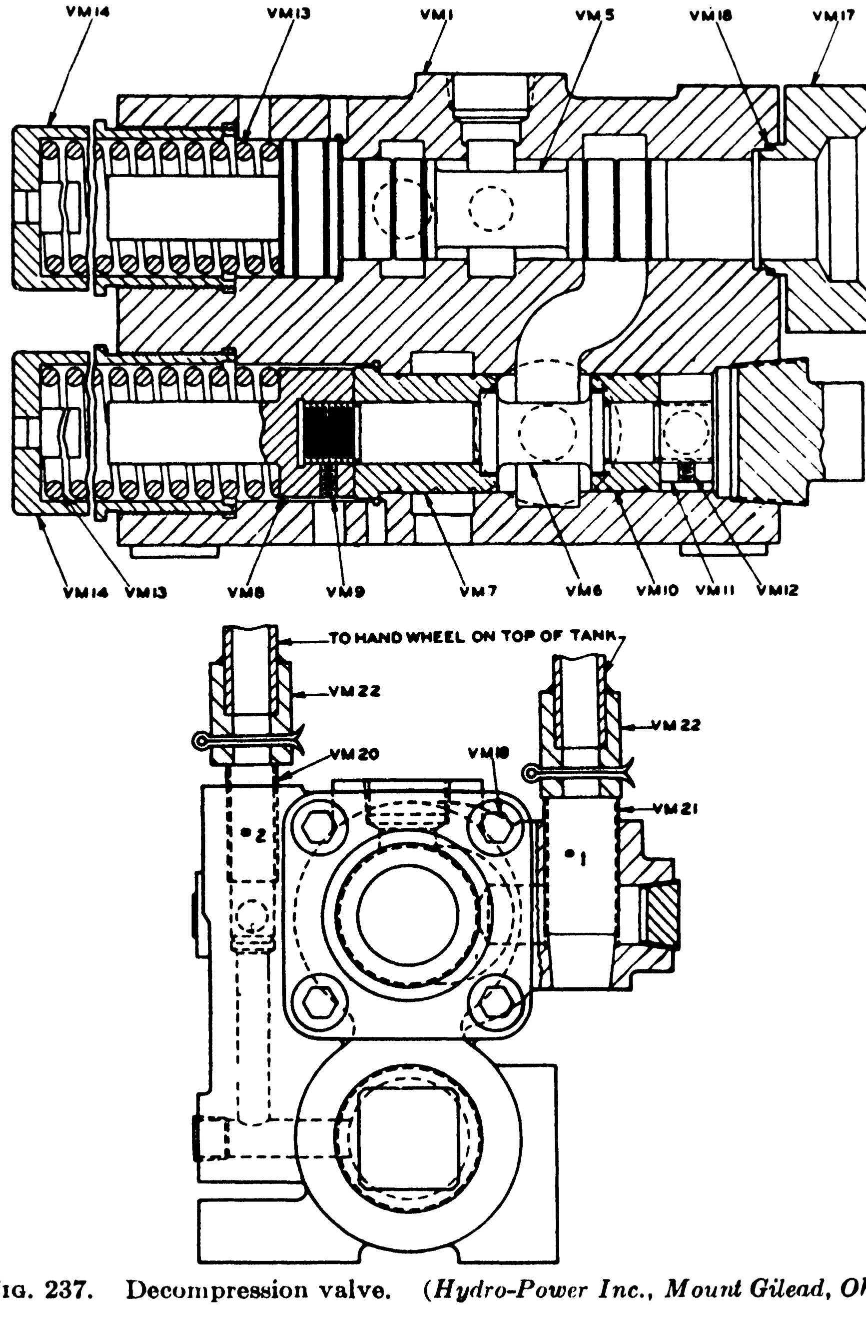


Fig. 237. Decompression valve. (*Hydro-Power Inc., Mount Gilead, Ohio.*)

case of failure of resistance. It should be noted also that in cases of differential areas, as in Fig. 235, the differential valve must be sufficiently large to permit make-up of oil from the tank for the head end of the piston, if descent takes place by gravity at the maximum rate possible by removal of oil from the differential area by the pump. Gravitational descent of a

piston in a reversing circuit may be prevented by back-pressure valves, in the same manner as in one-way circuits.

Prefill systems have been used extensively in connection with reversing pump circuits and form the basis of most modern high-speed oil-pressure presses. The circuit developed by the Oilgear Co. is shown in Fig. 238; it employs booster cylinders for rapid-traverse closing, a piston-type surge

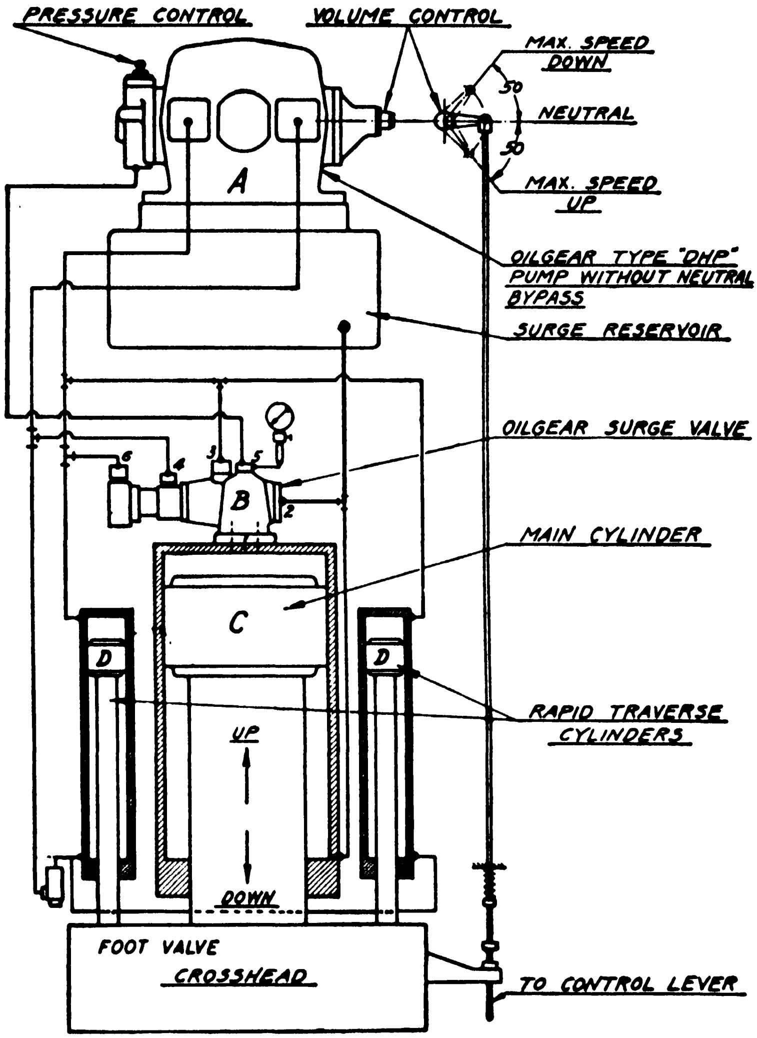


FIG. 238. High-speed-press oil circuit. (*The Oilgear Co., Milwaukee, Wis.*)

valve (see Fig. 186) with built-in sequence valve, and a standard servo-motor-controlled pump with integral two-way automatic suction-reversing valve (see Fig. 236). The diagram indicates a frequently used means to bring the ram to rest automatically at an adjustable point at the end of its return stroke. To this end a vertical rod is mounted in bearings fastened to the side of the press frame and connected to the servo-motor lever on the pump. The rod is forced to its lower position by a spring, shown in the illustration, mounted at its lower end and abutting against a bracket on the stationary part of the frame. An arm attached to the

traveling platen of the press may engage this rod by means of a stop collar fastened adjustably upon it and thus carry the rod with it at the time of engagement and actuate the servo-motor control so as to return the pump to its neutral or no-stroke position. The pump will automatically maintain the moving weights in their suspended position, with the pump on sufficient stroke to make up leakage losses. Raising the rod from its neutral position will cause discharge of the pump into the forward area of the cylinder and consequently downward movement of the press platen. Lowering the rod has the opposite effect. A control lever may be connected to this rod to effect control of the platen movement and speed manually. Intermediate stops may be accomplished by manually placing the rod in its neutral position.

To avoid a tendency to creep one way or the other, Oilgear pumps may be equipped with a neutral by-pass indicated at 5 in Fig. 236, which short-circuits the pump in dead neutral. If this device is employed, a resistance valve may have to be installed in the pullback line to prevent gravitational descent in neutral position of the pump.

Oil circuits similar to or identical with Fig. 238 are found in a number of makes of oil hydraulic presses presently on the market, although some companies have gone their own ways and developed original modifications of the reversing pump circuit to suit their particular purposes.

Automatic Operation of Reversing Pump Circuits. Automatic operation of reversing pump circuits has become common, particularly in the field of oil hydraulic machine tools and presses. Actuation by hydraulic pilot valves has almost completely superseded all other means of automatic control, and description of automatic controls will be confined to this mode of operation. Hydraulic controls developed for these pumps and described in Sec. 6, Chap. VII are utilized for this purpose.

Oilgear pumps are offered with several different hydraulically actuated controls, which may be operated by means of pilot valves to produce any desired hydraulic function. Type DM Oilgear pumps are supplied with hydraulic pilot cylinders permitting the selection of five positions, such as described in Chap. VII. A built-in multiple-head pilot valve selects these positions and may be actuated by cams or dogs on the table of the machine tool. Type DX pumps have three positions: forward, neutral, and reverse. The three positions are selected by means of a solenoid-operated pilot valve, which may be controlled to produce automatic action of a machine tool or press. The control operates in principle as shown in Fig. 101, and a double-solenoid-actuated pilot valve is required. One solenoid, when energized, produces forward movement, the other reverse. Deenergizing both solenoids puts the pump on neutral. Successful functioning of this system hinges on the installation of a back-

pressure valve when used with vertical cylinder, as it is difficult to put the pump on dead neutral, and a by-pass should be provided at the neutral position of the pump. An electric circuit, as shown in Fig. 228, may be set up to control the solenoid pilot valve and produce the desired automatic function.

This control may be applied to the pump in Fig. 238 to produce an automatic press. The control rod and hand lever shown in Fig. 238 would be left off and the pump completely controlled by push buttons and switches. Inching buttons may be provided for manual control of this press, if desired.

7. Multiple-hydraulic-motor Circuits. In the preceding sections of this chapter we have dealt exclusively with single pump-motor combinations; in other words, only one moving member on a machine had to be actuated and controlled, deriving its source of power from a suitably chosen hydraulic pump. However, many hydraulically actuated machines require power and control for a number of movements, generally in a predetermined sequence. In such cases, two alternate solutions present themselves, which have been briefly touched upon at the beginning of this chapter. In some cases, it will be advisable to provide an independent pumping unit for each of the motor units. When this is done, circuit design reverts to that discussed in preceding sections of this chapter, each individual pump-motor circuit being laid out to fit the particular requirements of its function. Tie-up by electrical controls may be made between the circuits to ensure maintenance of any sequence of functions that may be desired. The advantages of this arrangement are that greater flexibility is afforded in pressure and volume of the individual functions, that individual pumping units may be exactly adapted to individual motor units, and that there is considerably less complexity of valving and piping. The last-named characteristic is often decisive in the selection of the mode of operation, even though provision of several individual pumping units will be found more costly than a central source of hydraulic power.

As a general guide it may be said that on applications where there are one or two main functions performed at high pressure and speed, as, for instance, in oil hydraulic presses, die-casting machines, and heavy machine tools, it will be preferable to separate the main functions and handle them by individual high-pressure pumping units. This is particularly true when accurate control of speed and pressure is required, which may best be accomplished by adjustment of pressure and volume on the pumps themselves. Reversing or two-way pump systems should definitely be restricted to one pump and motor combination for each function.

For the average run of medium-pressure moderate-volume hydraulic systems, a number of schemes have been developed to obtain a series of motor functions from one pump; these will be discussed in the following.

a. Accumulator and Pressure-compensating Control Systems (Constant-pressure System). In this system of operation a constant pressure is maintained on the entire hydraulic system either by an accumulator (Sec. 6, Chap. X) or by a variable-delivery pump with pressure-compensating control. Accumulators may be operated in connection with variable-delivery pumps and pressure-compensating control or with constant-delivery pumps and unloading valves.

Accumulators of the elastic-load type are frequently used with forging machines, die-casting, and plastic machines and other machinery where large demands are made during short intervals, followed by longer periods of relative inactivity. Efficiency of this type of system is low, owing to the fact that the pressure on the system is constant, regardless of work resistance, and the pump is called upon to supply oil against this pressure continuously, except during by-pass periods.

In setting up a circuit for this type of operation, care must be taken to guard against loss of pressure fluid. Operating valves must be of blocked-center type (Fig. 166). Meter-in valves are indicated, if speed control of motors or cylinders is desired; otherwise speed of these devices will be governed by their work load and decrease as the load increases. Reducing valves may be installed to limit the pressure applied to any one hydraulic motor. If desired, both devices may be installed in series to control both speed and pressure of motor devices.

Variable-delivery pumps with pressure-compensating controls may be used for operation of multiple-motor circuits. The control will cause the pump to go on stroke and deliver oil when the pressure in the system drops on demand of cylinders or rotary motors. When demand ceases, pressure will build up and reduce the stroke of the pump to that required to maintain pressure in the system. Thus the pump will operate against pressure all the time, even though at reduced or practically zero delivery part of the time. The efficiency of this mode of operation is good, if the pump is connected to the operating valves without intervening reducing or metering valves, so that its discharge is not throttled and pressure may fluctuate with work load.

There are many cases where a piston must be brought forward and held under pressure, following which an operation is performed on a second cylinder. If this second cylinder requires a lower pressure, then the pressure may drift out of the first or holding cylinder, even if a check valve is placed in the line ahead of it, owing to slippage losses past valve pistons and piston rings. To prevent this, a meter-in valve may be used

ahead of the second cylinder, so that not all the pump or accumulator output will be used to operate it and a head of pressure is maintained on the entire system. A separate small holding pump is sometimes used to accomplish the same result.

b. Operation with Sequence Valves. Where a specific predetermined sequence of hydraulic movements is desired, sequence valves may be

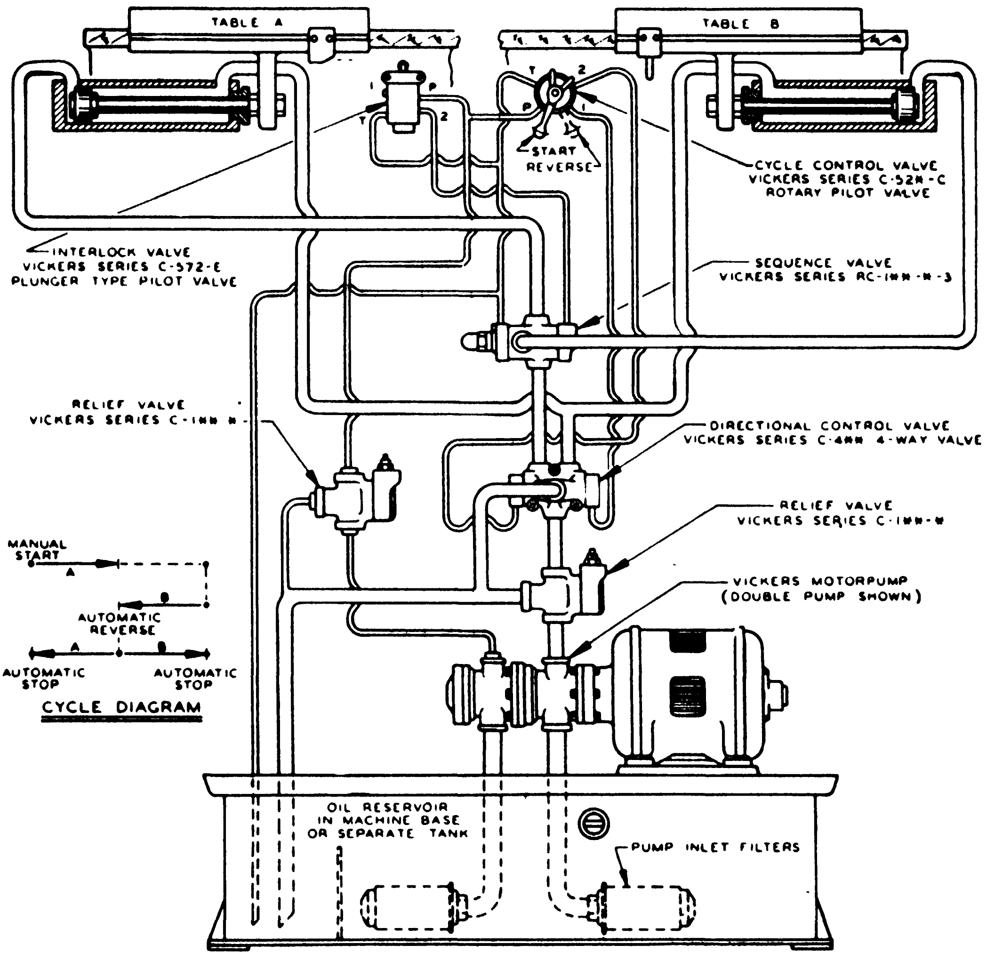


FIG. 239. Hydraulic circuit with pilot-valve-controlled sequence valves. (*Vickers Inc., Detroit, Mich.*)

employed to advantage. These valves, described in the preceding chapter, ensure the build-up of a predetermined pressure in a given cylinder before permitting application of hydraulic power to the succeeding cylinder. An application showing sequence valves operated by pilot-valve-controlled pressure is shown in Fig. 239. In this circuit pressure is first applied to cylinder 1 by a pilot-operated four-way valve, so that the ram may advance until the pilot valve is engaged, which admits pressure to the corresponding sequence valve, permitting the second ram to advance. Reversal of pilot valve causes reversal and simultaneous retraction of all cylinders through check valve built into

the sequence valve. No provision is shown in this diagram to unload or by-pass the pump at the end of the return stroke; this may be easily done by any of the means previously described.

If several cylinders or hydraulic motors are operated from a common source of hydraulic power and pressure is applied to them simultaneously, then the speed of movement of the individual cylinders will, of course, depend on their resistance. It is a practical impossibility to obtain uniform movement of two or more cylinders connected to the same power

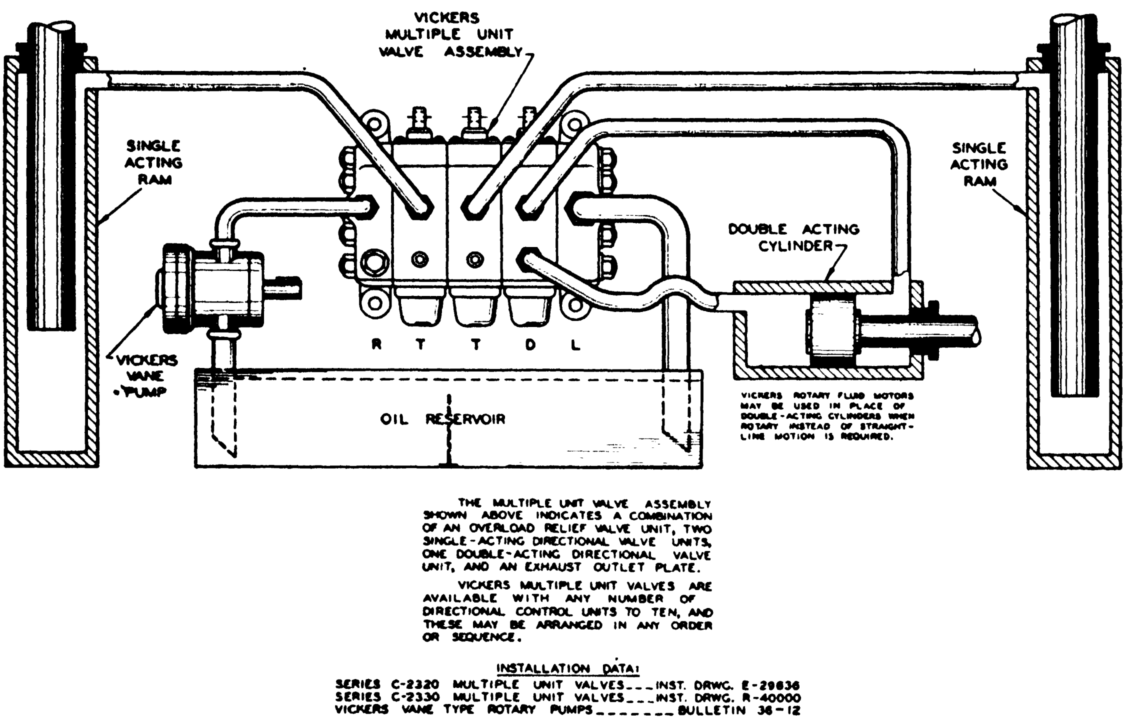


FIG. 240. Hydraulic circuit with multiple control valve. (*Vickers Inc., Detroit, Mich.*)

source, no matter how carefully the loads may be equalized. The same reasoning applies to the beam of press brakes and similar devices. Attempts have been made to force uniformity of movement by application of the different kinds of metering valves and locked circuits previously described or by flow dividers (see Sec. 4, Chap. X). None of these attempts have been 100 per cent successful, owing to variation in slippage and oil compressibility and to malfunctioning of flow-control devices. The designer of hydraulic machinery will do well to accept this fact and provide mechanical means to tie the hydraulic rams together so that they will be restrained mechanically from getting out of time.

c. The Open-center Hydraulic Circuit. This circuit has made a humble beginning in agricultural and road-machinery applications, but is becoming increasingly popular because of a number of advantages. It is based on the design of the center-by-pass valve shown in Fig. 174. Any num-

ber of these valves may be put in series to form a "sandwich" and to control any desired number of hydraulic functions. With all valve pistons in neutral position, a free by-pass is provided for the pump, which discharges into the supply tank at practically zero pressure. As soon as any one of the valve spools is shifted, pressure is applied to the corresponding device without disturbing or affecting any other device. All cylinders not in action are locked by the valve pistons covering their respective ports. A circuit incorporating the series center-by-pass valve is shown in Fig. 240.

8. Hydraulic-electric Analogy. No discussion of hydraulic circuits would be complete without description of the hydraulic-electric-analogy method developed by Hans Ernst, Director of Research of the Cincinnati Milling Machine Co. Considerable information on this ingenious and convenient means of circuit analysis has been given in articles by Ernst and Albert H. Dall, which appeared in different trade magazines[1] and are utilized in this text with permission and through courtesy of the Cincinnati Milling Machine Co., Cincinnati, Ohio.

The analogy is based on the similarity of viscous fluid flow to that of direct electric current, the one governed by Poiseuille's equation, the other by the well-known Ohm's law. Of course, not all flow is viscous flow, but for the purpose of this discussion we assume that it is, and this somewhat arbitrary assumption has led to the discovery of many very interesting devices, which might not have been discovered otherwise.

From Eq. (7), Chap. V, we obtain

$$p = \frac{128\mu LQ}{\pi D^4} \tag{7}$$

where p denotes the pressure drop across the pipe of diameter D and length L. In this equation all quantities are expressed in fundamental units of the English gravitational system. Expressing p in pounds per square inch, L and D in inches, and Q in cubic inches per minute, we have

$$p = \frac{0.0047\mu LQ}{D^4} \tag{8}$$

or

$$\frac{p}{Q} = \frac{0.0047\mu L}{D^4} \tag{9}$$

The expression on the right side of Eq. (9) is a quantity characteristic for

[1] Hans Ernst, Modern Hydraulic Control Systems and Circuits, *Product Eng.*, April, 1935, 121; May, 1935, 171. A. H. Dall, Machine Hydraulics, *Machine Design*, April, 1946, 143; June, 1946, 111; August, 1946, 125; October, 1946, 87; December, 1946, 119; Hydraulic Circuits, *Product Eng.*, August, 1939, 336.

the physical dimensions of the pipe and the properties of the fluid passing through it. If we designate this expression as R, we may write

$$\frac{p}{Q} = R \tag{10}$$

and obtain an analogy to the well-known Ohm's law of electrical circuits, $E = IR$.

Resistances in hydraulic circuits may be made from choke coils or calibrated pins in fixed orifices. The pressure drop across these chokes may be utilized in a hydraulic circuit to operate pilot or control valves, just as the voltage drop across an electrical resistance is used to send current through a circuit shunted around it. Combination of hydraulic resistances in series and parallel may be analyzed in precisely the same manner as similar combinations of electrical resistances.

To illustrate, we will consider two parallel circuit branches, having the resistances R_1 and R_2, respectively, and Q_1 and Q_2, the corresponding rates of flow. We have

$$Q_1R_1 = Q_2R_2 = p_0 = (Q_1 + Q_2)R \tag{11}$$

where p_0 = pressure drop across resistances
R = total resistance

From Eq. (11) follows

$$R = \frac{R_1R_2}{R_1 + R_2} \tag{12}$$

resembling the well-known resistance formula for two parallel electrical branches, and

$$p_0 = \frac{QR_1R_2}{R_1 + R_2} \tag{13}$$

It is interesting to compare this equation to that developed for the pressure drop in divided flow in Sec. 6, Chap. VI. The latter [Eq. (29), Chap. VI] is based on turbulent flow. If two resistances are placed in series, we may take off a header line at an intermediate point and thus obtain a pressure p_1 that will always be a fixed fraction of the input pressure p_0. If the resistances are designated as R_1 and R_2, then p_1 will be

$$p_1 = p_0\left(\frac{R_2}{R_1 + R_2}\right) \tag{14}$$

This principle is utilized in the "hydraulic potentiometer," where resistances R_1 and R_2 are changed in a complementary fashion to obtain

equated. Therefore

$$\frac{Q_1}{Q_2} = \frac{R_2}{R_1} \tag{17}$$

Flows Q_1 and Q_2 are independent of the pressures in the system. If resistances R_1 and R_2 are made exactly equal, the flows in each leg of the bridge will be equal, regardless of loads L_1 and L_2, either one of which may be zero.

Other hydraulic devices and appliances may be analyzed in a similar manner by means of this convenient and ingenious method.

CHAPTER XII

INDUSTRIAL APPLICATIONS OF OIL HYDRAULIC POWER

1. General Industrial Applications. The concluding chapter of this text will be devoted to the description of hydraulically actuated and controlled machines built by some of the leaders in the field of applied hydraulics. The field of hydraulic applications has become tremendous, and an attempt has been made to show and describe only a very few of the most interesting ones to acquaint the reader with what has been accomplished and the possibilities existing for further development.

In the preceding chapter the author has touched upon the general fields in which hydraulics could be used to advantage, and a check list prepared by the Oilgear Co., Milwaukee, was reproduced to help the designing engineer decide whether employment of this means of power and control was indicated. In this chapter we shall deal with applications that have actually been made and that operate successfully. As the title of this book implies, only industrial applications of hydraulics are dealt with. These cover mainly hydraulic drives and controls for machine tools, such as milling machines, grinders, planers and shapers, drilling and boring machines, and other standard and special machine tools. Another large field is use of hydraulics in oil hydraulic presses, die-casting and plastic machines, and other process machinery. A few representative examples of each category are given in the following.

2. The Oil Hydraulic Press. One of the most outstanding applications of oil hydraulic power has been made in connection with modern oil-pressure presses. The hydraulic press has always constituted one of the most useful tools in the metal-working and processing industry, and its wider application has been prevented only by its inherently slow speed and cumbersome control. Advent of oil hydraulics with its speed and ease of control precipitated a rapid development that culminated in the modern oil hydraulic press, now offered by a number of manufacturers in a variety of styles and sizes, supplied with automatic controls, permitting any desired mode of operation, and capable of production speeds that in many cases exceed those of competitive mechanical equipment. Hydraulic circuits of great complexity have been developed to permit many of the complicated operations demanded by the processing and metal-working industries.

A typical application of hydraulic power and control is shown in the triple-acting forming and drawing press made by the Hydraulic Press Mfg. Co. (Fig. 243). This machine is equipped with a main punch or draw ram of the telescoping type, having an internal or stationary ram for the first part of the draw stroke, which may be made at high speed and low tonnage. After the initial part of the draw stroke has been completed, pressure will rise and open a sequence valve directing the flow of oil into the large area of the outer ram to complete the operation at slower speed and higher tonnage. During the operation, the blank being drawn is supported by a blank holder that securely clamps it against the dies, so that no wrinkles will develop as the blank is pulled out between blank holder and die and drawn into the finished shape. The lower part of the piece being drawn is subjected to very heavy pressure by the punch and must be prevented from rupturing by supporting it with an auxiliary hydraulic ram, mounted in the bed of the press, the so-called "die-cushion" ram. Rapid traverse to and from the work is provided as well as full automatic control by electric push buttons and pressure switch for automatic reversal at a preset pressure limit. Presses of this kind are supplied in any combination of tonnage, stroke, and bed size and powered with variable-delivery pumping units ranging from 25- to over 400-hp capacity, according to size and speed desired. The hydraulic system of this press is shown in Fig. 244 and will be described in the following.

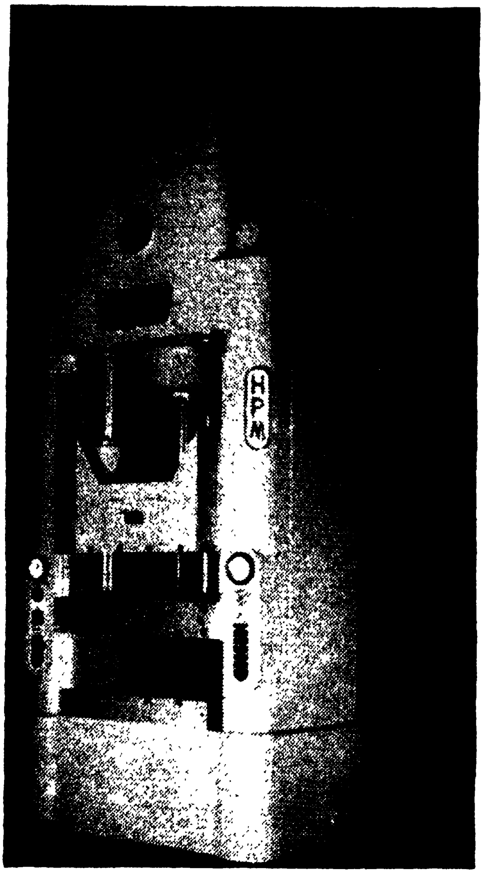


Fig. 243. Triple-acting oil hydraulic drawing press. (*The Hydraulic Press Mfg. Co., Mount Gilead, Ohio.*)

Power is supplied to the main or punch-operating cylinder by a variable-delivery reversible-discharge pump connected in a two-way system. High-pressure pipe lines lead from the pump connections to the internal area of the stationary booster ram and to the retraction area of the main ram. A purely gravity rapid-traverse system with prefill check valves is employed. A decompression valve permits decompression of the oil in the press cylinder at a controlled rate, simultaneously permitting the pump to by-pass its output until the pressure has been fully

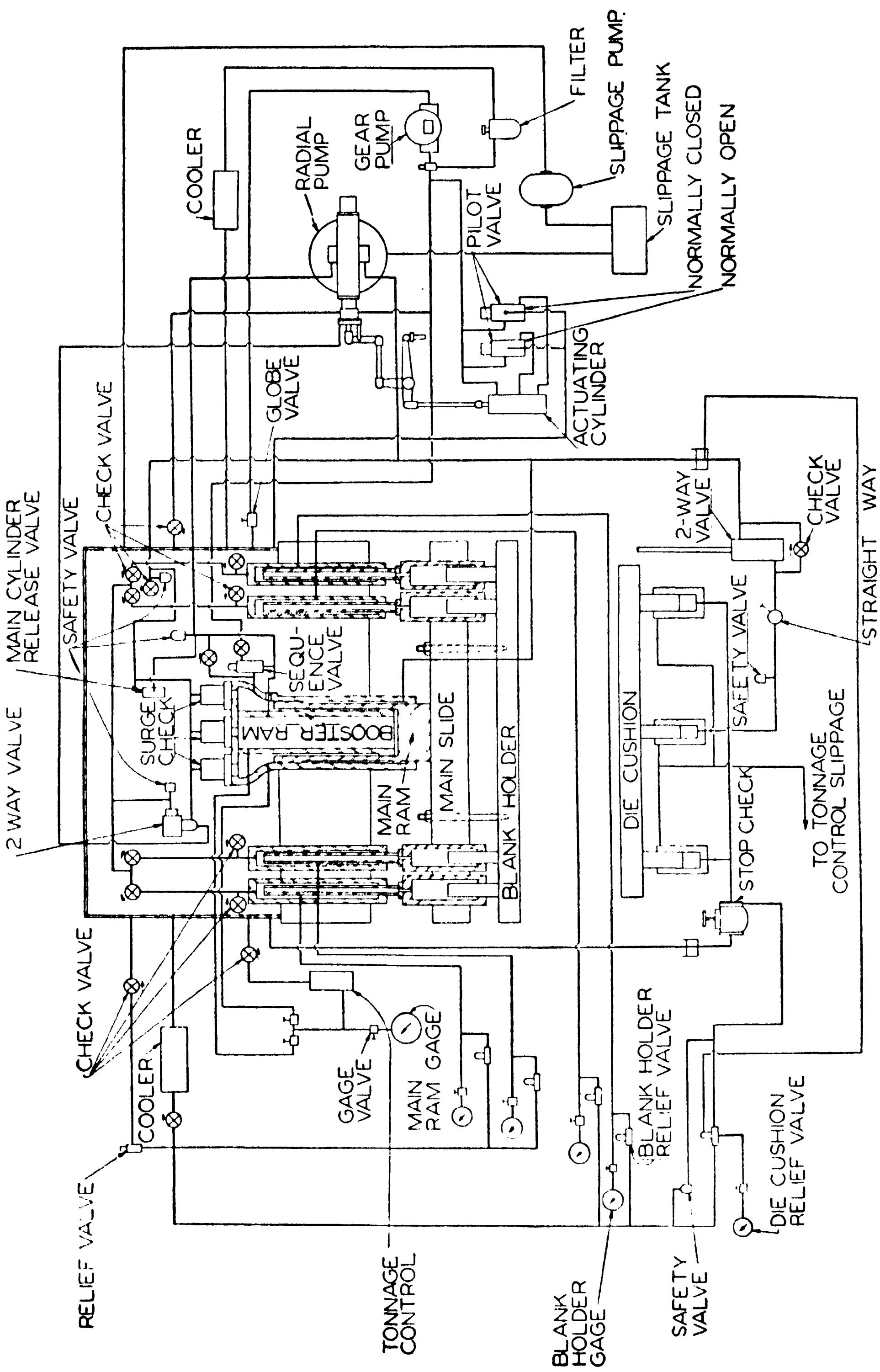


FIG. 244. Hydraulic circuit of triple-acting hydraulic draw press. (*The Hydraulic Press Mfg. Co., Mount Gilead, Ohio.*)

released in the press cylinder, and the platen may be returned by the pump discharging into the pullback area, after forcing open the prefill check valves by means of the pilot pistons. Control of direction and speed of the press platen is accomplished in the manner described below.

A servo-motor control (see Fig. 109), which governs magnitude and direction of pump output, is actuated by bell crank and linkage through a control rod and hand lever. The circuit operates as follows: We assume that the moving platen is suspended in its adjustable top position by the engagement of its control arm with the vertical control rod. The pump is on neutral stroke, discharging just enough oil to keep the moving parts suspended and to make up leakage losses within itself and the system. The line from the pump to the upper end of the main cylinder is under suction, and the pump may draw oil from the supply tank through the surge check. The decompression valve, shown in section in Fig. 237, will have both its pistons closed, unless the static pressure created by the action of the moving weights on the pullback area is sufficient to open the upper one of the two plungers. This will not cause any action, as the lower or by-pass piston is, of necessity, closed, there being no pressure in the pressing cylinder cavity. If now the servo control of the pump is shifted to the "forward" position, the pump will take in oil from the pullback line and discharge into the pressing line. The platen will descend rapidly at a speed depending upon the rate of withdrawal of oil from the pullback area. The surge check valves will open, owing to the vacuum created in the cylinder by the rapid descent of the platen, and prefill the cylinders with oil. At the same time the pump continues to discharge into the cylinder space above the piston. After the platen meets with resistance and its descent is checked, the surge check valves will close. The pump continues to discharge into the cylinder and will now draw oil from the supply tank through the check valve provided for this purpose. Pressure begins to build up and is transmitted to the decompression valve through a pipe line provided for this purpose. The pump is now reversed, and the cylinder pressure line becomes suction, and the pullback line becomes discharge. Pressure applied to the pullback line will force open the upper of the two pistons in the decompression valve, opening the pressure to the exhaust through an adjustable restriction marked 1 in Fig. 237. As pressure from the cylinder bleeds out at a controlled rate, the pump discharge passes from the upper passage through the diagonal port through the open lower valve into the tank. This action of the pump assists in decompressing the cylinder, but it may readily be seen that the pump is not obliged to perform this service alone, so that decompression will take place at a much faster, though fully controlled rate. After pressure in the cylinder has dropped down to the

setting of the spring in the lower of the two valve plungers, this plunger will close at a rate controlled by choke 2, which may be so set as to ensure a complete and full release of all pressure, so that a smooth and absolutely shockless reversal may be accomplished. This will take place as soon as the lower piston has closed off the by-pass port, permitting the pump to apply its discharge to the pullback area, force open the surge checks by means of the pilot plungers, and return the moving platen to its initial starting position. Mention should be made of the fact that care must be taken to dimension the pilot plunger on the surge valve so that the valve is sure to open under all conditions. It may be seen readily that a hydrostatic relationship exists that may be expressed as follows:

$$\frac{p_R A_R}{A_F} a_C < p_R a_P \quad \text{(neglecting moving weights)} \qquad (1)$$

where p_R = retraction pressure
A_R = retraction area
A_F = forward area
a_C = surge check area
a_P = pilot plunger area

From this follows

$$a_P > \frac{a_C A_R}{A_F} \qquad (2)$$

A pilot-valve-actuated piston connected to the servo valve is used to achieve a series of automatic functions in their desired sequence. Installation of the hydraulic cylinder, or control operator, connected to the servo-valve lever, is shown in Fig. 245. The servo-valve lever, actuated by the control rod, may be seen immediately in back of the control-operator rocker arm and is connected to the rocker shaft by a lost-motion cam so that movement of the lever will cause shifting of the servo valve towards neutral, but action of the control operator will not cause movement of the lever, in order to minimize the inertia of the parts moved by the control operator.

The principle of the control operator and its operation is illustrated in Fig. 246. In the particular cylinder shown, the stroke has been made 2½ in. total, and the ratio of lever arms on the rocker is made so that this 2½-in. stroke on the operator will produce movement from hard over to hard over on the servo valve and pump shift ring. Valve 1 controls the reciprocation of the control and with it that of the press. In the position shown, this valve is energized, connecting line FR to the exhaust. Pressure on top of piston through line E has brought piston A to its lowest position, corresponding to full pump stroke on "down" or "advance" position.

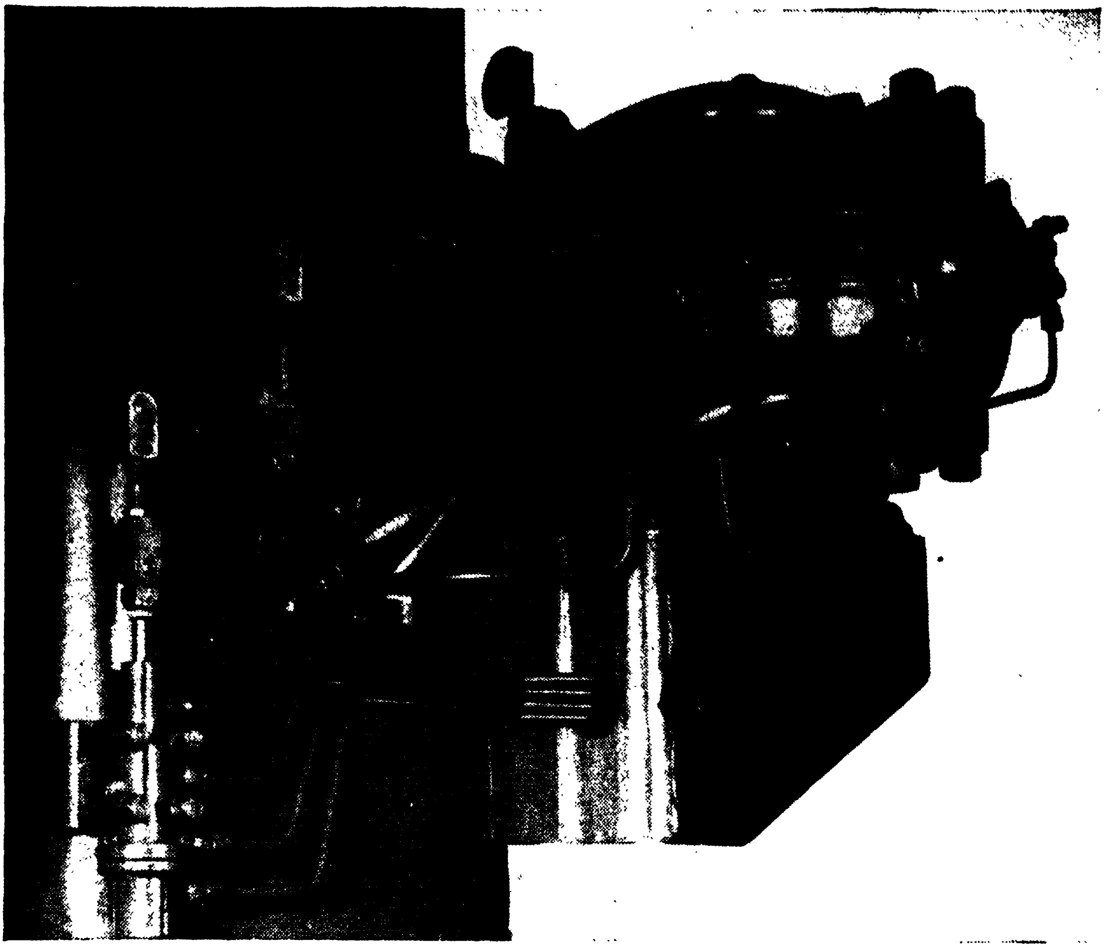


FIG. 245. Hydraulic control operator. (*The Hydraulic Press Mfg. Co., Mount Gilead, Ohio.*)

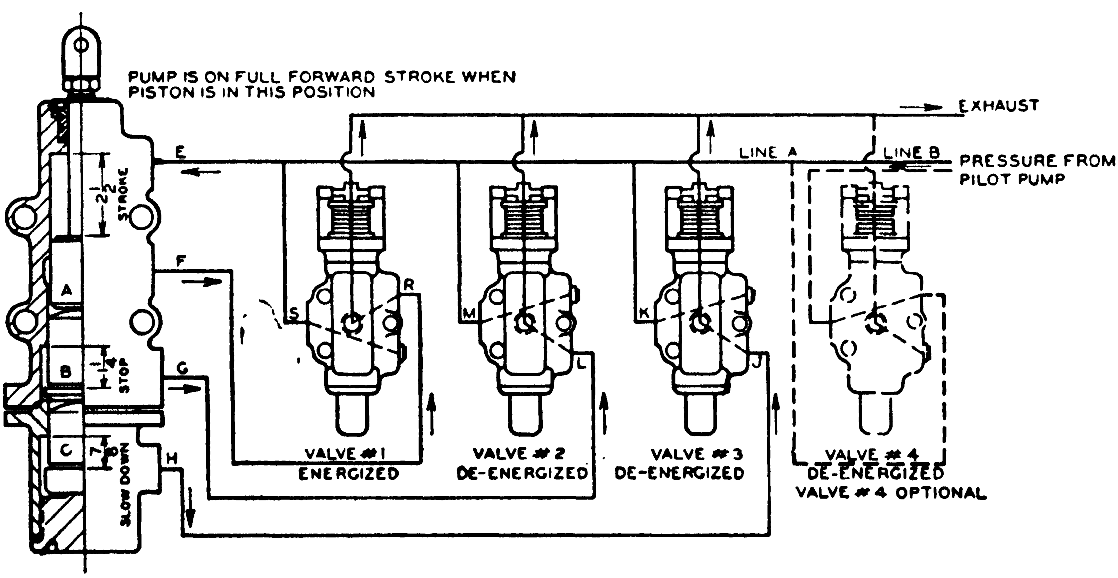


FIG. 246. Hydraulic control operator and pilot valves. (*The Hydraulic Press Mfg. Co., Mount Gilead, Ohio.*)

Valve 2 is the "stop" valve. If this valve is energized at any time during the down stroke, pressure will be applied to piston B, which, by virtue of its larger area, will force piston A toward the neutral or no-stroke position, which is reached when a shoulder on B engages a stop in the housing, having made a 1¼-in. stroke. By provision of a suitable "stop" button, valve 2 may be energized at any point of the working stroke and the press platen stopped (subject to a slight amount of creeping due to inability to reach dead neutral position). Valve 3 is the "slow-down" valve. Energizing this valve by limit switch or push button causes piston C to travel ⅞ in., causing a reduction in pump stroke and with it a reduction in pump output and press speed. Deenergizing this valve causes the press to resume speed. The purpose of valve 4 is to relieve pressure in the operator to enable control of the press by hand-lever manipulation, as previously described. Deenergizing valve 1 by a pressure- or position-controlled switch causes pressure to be applied to line FR, which moves piston A all the way up 2½ in., thus putting the pump on hard over reverse. The platen will then be retracted until the stop collar engages the control rod, which in turn actuates the control lever forcing the servo valve to neutral position.

In laying out a control of this kind, consideration should be given to the speed at which the servo valve is moved, which should not exceed the rate at which the servo piston can follow. The length of the servo-valve lever, therefore, should be computed so that the linear speed of the control rod, which equals that of the press platen, will produce a linear speed of servo-valve displacement less than that at which the servo piston travels when propelled at the rated capacity of the auxiliary pump.

The servo valve is moved against the force of the pressure acting on plunger A, which continuously tends to force it on "return" position, but the force available to actuate the control rod by far exceeds this resistance. The operator, however, is not capable of exerting sufficient force to do this, and for that reason valve 4 has been provided. An electric control circuit is provided to permit actuation of the solenoid valves in the desired sequence. Suitable push buttons and selector switches, conveniently arranged, permit selection of the desired functions.

Hydraulic power is supplied to both servo motor and actuating cylinder by an auxiliary pump of the gear type, equipped with relief valve. This pump is utilized to operate an oil cooler, diagrammatically shown in Fig. 244. A slippage or scavenging pump is provided to return leakage oil from a sump beneath the main pump to the overhead tank. An auxiliary line may be observed running from the pilot-pump discharge to the sequence valve in the tank. The purpose of this pressure supply is to "reset" the sequence valve, which is pilot controlled with a pilot relief

valve opening at a preset pressure and permitting the plunger to open and admit pump pressure to the main ram area. To facilitate release from the main cylinder, the sequence valve is shunted by a check valve.

We shall now describe the means by which pressure is supplied to the blank holder that clamps the blank during the drawing operation. For this purpose, the main or punch slide, operated by the main ram, has mounted in it four hydraulic cylinders, which actually are spaced at the four corners, but are shown in pairs adjacent to each other in Fig. 244 for the sake of clarity. A blank-holder slide is suspended directly below the punch slide by means of adjustable pickup rods, as indicated in the diagram. Rams are fastened to this blank-holder slide and extend into the cylinders on the main slide. Hollow rams of smaller diameter are mounted upon the cylinders and extend into secondary cylinders, which are fastened to the head of the press. In operation, the entire main and blank-holder-slide assembly will descend at rapid-traverse speed until the blank holder engages the blank, tending to prevent any further movement. During this advance stroke, oil will be drawn into the stationary head cylinders through check valves provided for this purpose. As the blank holder is prevented from continuing its downward travel, oil is trapped in the blank-holder cylinder assemblies, which sets up a resistance toward further movement, so that rapid-traverse movement of the slide will cease and pressure will rise in the main cylinder, while the main surge valves close. Oil will be forced out of the blank-holder cylinder through relief valves set for the desired blank-holder pressure. Each cylinder has a separate relief valve so that independent pressures may be set up at the four corners, whereby a blank of unsymmetrical configuration may be clamped with pressures varying around the contour. Pressures developed on these blank-holder cylinders are indicated on separate gauges, and a record may be made after a certain die has been run so that the valves may be quickly reset in case of a future rerun. It may be seen that the pressure exerted by the blank holder equals that of the internal hydraulic pressure times the area of the lower blank-holder rams. The purpose of the ram extensions is to balance this load partly, so that the main ram will not be called upon to overcome all the blank-holder resistance during the draw stroke, which would detract from its usable working capacity and lower the efficiency of the machine. To this end, the area of these balancing rams is made from one-half to two-thirds that of the blank-holder rams, so that the drawing ram must overcome only from one-third to one-half of the blank-holder load. Oil discharged from the blank-holder-cylinder relief valves is collected in a common header and passed through an auxiliary cooler into the oil tank.

The action of the die cushion is somewhat similar to that of the blank-

holder, except that the die cushion will support the section of the blank immediately under the center of the punch, particularly in cases where the punch is pointed, as in the case of automobile headlights or similar bullet-shaped objects. This is done by pins extending through the bolster plate into the die, which support a movable section of the die bottom and about the die-cushion platen mounted in the bed of the press underneath the bolster plate. At the beginning of the drawing operation, this die-cushion platen is elevated against the bolster plate, and the punch support is approximately level with the blank placed upon the die. As the draw punch engages the blank, the punch support will prevent the punch from rupturing the blank and will yield at a preset pressure existing in the two outer die-cushion cylinders determined by the die-cushion relief valve. Oil displaced by these cylinders escapes from the relief valve into the common header.

After the completion of the draw stroke, a preset pressure is built up in the system and causes the pressure switch to act and reverse the stroke of the pump and with it the movement of the punch ram, as previously described. When this is done, a branch connected to the pullback line will open a pilot-actuated two-way valve to release pressure from the blank-holder cylinders so that the assembly may be retracted. All blank-holder cylinders have exhaust lines connected to a common header via individual check valves, and the common header may be connected to the tank by the pilot-operated two-way valve. The purpose of the individual check valves is to prevent equalization of pressure between the individual cylinders when on pressure stroke. A branch line connected to the pilot-operated two-way valve supplies pilot pressure to an auxiliary piston on this valve, which opposes the action of the pullback-pressure-operated pilot piston. Pilot pressure is supplied on down stroke only, when the servo valve in the pump control applies pilot pressure to the forward end of the control to which the pilot line is connected. Thus on down stroke, the two-way valve is held firmly closed against any possibility of being forced open by the pullback pressure created by the weight of the moving parts. A similar action takes place in the die cushion. A branch line from the pullback connection leads to a two-way valve that is operated from the press slide by a suitable linkage rod. After the punch has been withdrawn from the finished piece, it is necessary to retract the blank holder first to prevent damage to the piece by the punch support trying to eject it from the die. Therefore, admission to the central die-cushion-lift ram is delayed until the blank holder has withdrawn to clear the piece being ejected from the die. Pressure will then be admitted to the die-cushion-lift ram in center to return the die cushion to its elevated position and eject the piece. During this period, the

outer cylinders are prefilled through a check valve from the overhead tank. At the end of the retraction stroke, the slide will come to rest by engaging the pump control.

3. Milling Machines. *The Cincinnati Milling Machine Co.* This company has become one of the outstanding leaders in the field of oil hydraulics, in both theoretical development and practical application. Best known in their line of production milling machines is the Hydromatic milling machine, illustrated in Fig. 247, which represents one of the

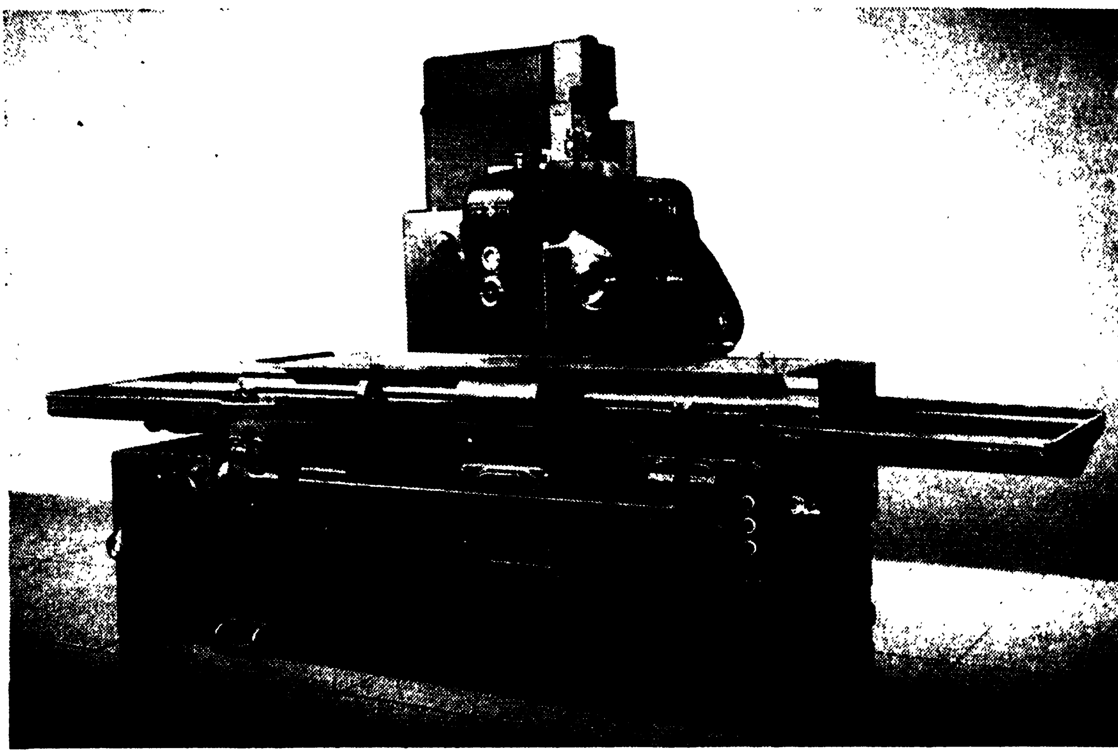


FIG. 247. Cincinnati Hydromatic milling machine. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

pioneering achievements of this company. The machine is a bed-type production milling machine. The table is traversed by a hydraulic cylinder located underneath it, and a multiplicity of automatic and semi-automatic cycles may be had by the mere setting of dogs. The feed-rate control lever is located on the hydraulic unit at the left end of the bed, as seen in Fig. 247. Other levers on the bed control such functions as start-stop, feed, and traverse rates.

The hydraulic circuit of the machine is illustrated diagrammatically in Fig. 248. This is the original Cincinnati locked circuit with a variable-delivery pump interposed between back and forward pressure lines, the setting of which determines the feed rate. A very small capacity, so-called "booster," pump is connected to the forward pressure, which

maintains pressure on both forward and back pressure and makes up leakage losses in the variable-delivery pump and system. Relationship between forward and back pressure is determined by the differential relief valve, which ensures that the sum of back and forward pressures remains constant and permits a complementary rise and fall of both forward and back pressure to suit the work resistance. (For a full analysis of the differential-relief-valve action, refer to Sec. 3, Chap. XI.) An auxiliary large-capacity pump C is provided to permit rapid-traverse

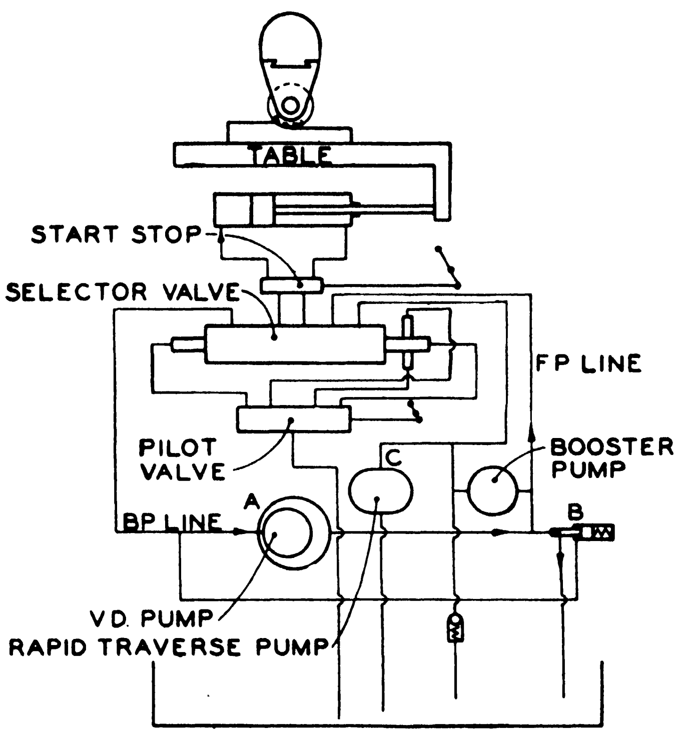


FIG. 248. Locked hydraulic circuit of Hydromatic milling machine. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

action in either direction. The selector valve permits selection of direction and rate of travel by a combination of rotary and oscillatory adjustments controlled by a pilot valve. A start-stop by-pass valve completes the circuit.

One of this company's most recent achievements is the tracer-controlled milling machine, used for duplicating and contour milling. The principle of the tracer control consists in the application of precision servo-motor controls, either as control or power servos. A brief description of servo controls was given in Sec. 6, Chap. VII. Servo motors covered in that chapter were control devices used to control stroke and delivery of radial variable-delivery pumps. In that case, the variable-delivery pumps were the main power source, and the servo was used to control the action of the power generator. Power servos, operating on a

similar principle, directly control the flow of primary power and in themselves are pilot-controlled hydraulic motors. In the following, the action of a power servo will be analyzed by means of the electric-hydraulic-analogy method (Sec. 8, Chap. XI).[1]

Figure 249 shows diagrammatically a simple type of power servo. Hydraulic power is supplied to the center piston in the servo valve, and movement of this valve in either direction will cause this pressure to be

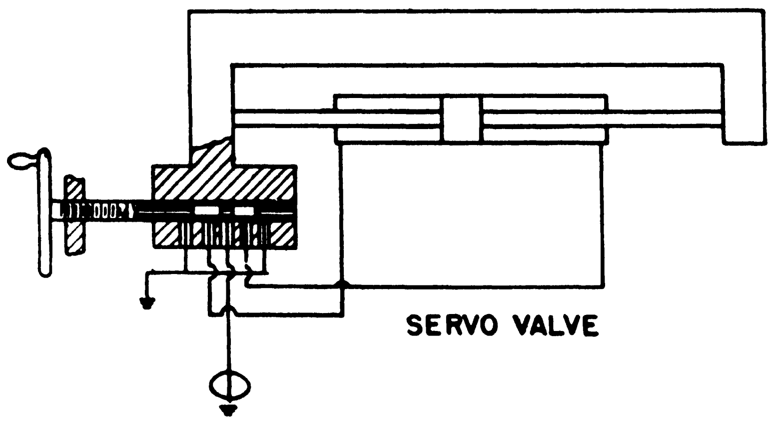


FIG. 249. Servo valve. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

applied to the corresponding end of the table-driving cylinder, while the opposite end is being exhausted. The neutral position is restored by the movement of the slide to which the servo-valve housing is rigidly fastened. Figure 250 shows how the action of this control may be analyzed. Each port and its corresponding valve shoulder may be considered as a resistance R, subject to the law of hydraulic flow, developed in the preceding chapter.

$$p = QR \tag{3}$$

For two resistances in series we have established the equations

$$p_1 = p_0 \frac{R_2}{R_1 + R_2} \tag{4}$$

and

$$p_2 = p_0 \frac{R_4}{R_3 + R_4} \tag{5}$$

[1] For further information refer to A. H. Dall, Sensitive Hydraulic Servos and Tracer Control System for Machine Tools, *Machine Design*, December, 1946. The analysis given above is taken from the reference article by permission of the Cincinnati Milling Machine Co., Cincinnati, Ohio.

Since $R_1 = R_4$ and $R_2 = R_3$ by design, we have

$$p_1 - p_2 = p_0 \frac{R_2 - R_1}{R_2 + R_1} \tag{6}$$

and

$$p_1 - p_2 = p_0 \frac{R_3 - R_4}{R_3 + R_4} \tag{7}$$

The term $p_1 - p_2$ is the difference in pressure at both sides of the power piston, which causes movement thereof. It may be seen very readily

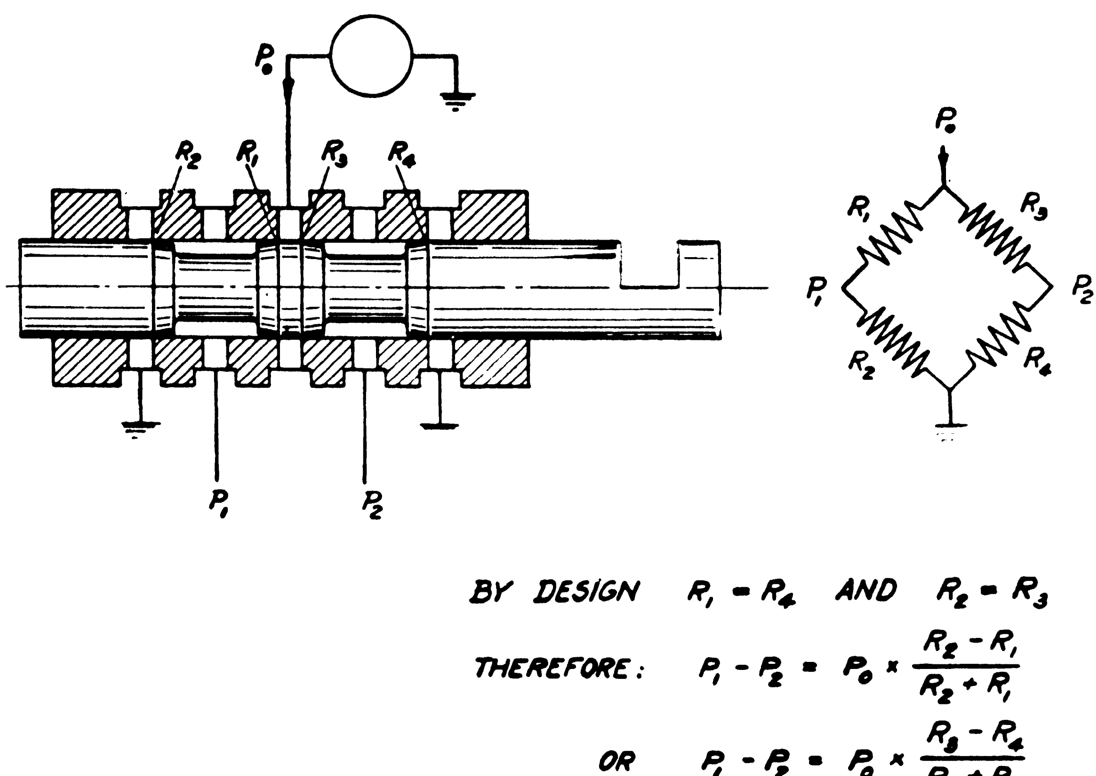


FIG. 250. Theory of servo valve. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

that for a very slight movement of the valve in either direction, R_2 or R_1 will become very large, so that $p_1 - p_2$ approaches $\pm p_0$. The disturbance of the pressure equilibrium produces both flow through the junctions and restoring movement of the slide. This causes a rapid decay of pressure difference so that the mechanism will again come to rest.

Great sensitivity may be had owing to the rapid increase of pressure difference with a slight movement of the valve. This produces both power to produce table thrust and flow to produce table movement. Curves made from actual test data, which were taken from Dall's article,

vividly illustrate this important characteristic. Figure 251 shows flow plotted against valve movement for both a supersensitive and a less sensitive valve. A similar graph in Fig. 252 illustrates the relationship between pressure difference and valve position.

Mention should be made, and it has been pointed out in the original article, that highly sensitive servos are given to the possibility of oscillation or "dither."

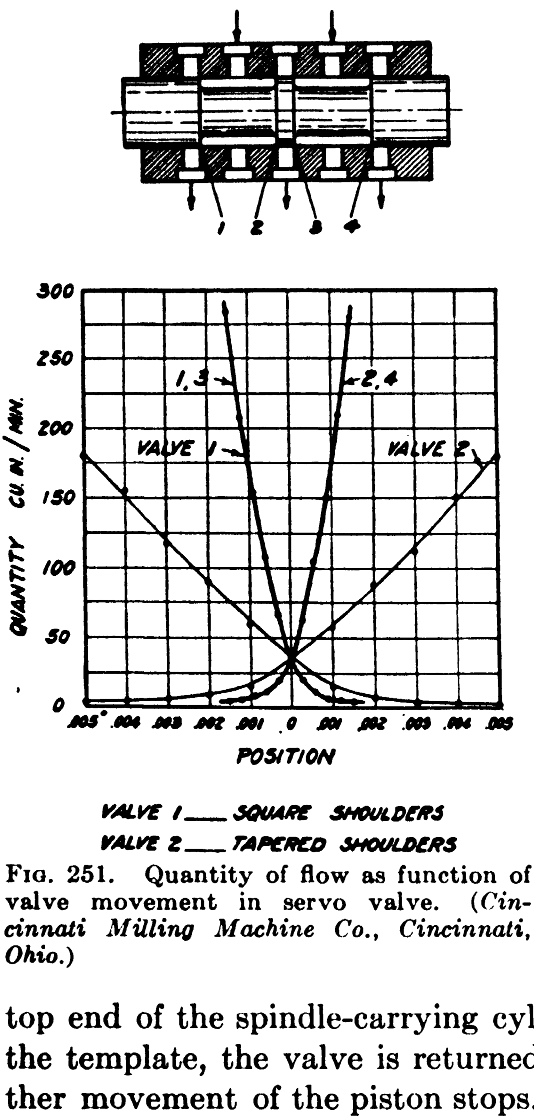


FIG. 251. Quantity of flow as function of valve movement in servo valve. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

In the following the application of servo controls to tracer mechanisms will be described. Tracer controls are available both for 180° (axis of the tracer in the plane of the contour) and 360° contour milling. Figure 253 shows the principle of a 180° tracer. Here the tracer controls the valve of a power servo, which in turn operates the hydraulic cylinder carrying the cutter head. The table that carries work and template is operated by another cylinder, which is also subject to the control of the servo valve. This is done by provision of a servo-valve-controlled throttling port, which controls the outlet from the back end of the table-propelling cylinder. The tracer seeks contact with the template, as the spring on top of the servo valve will force this valve to its maximum down position, admitting pressure to the top end of the spindle-carrying cylinder. As soon as the tracer engages the template, the valve is returned toward its neutral position, and further movement of the piston stops. At the same time, the throttle port on top the valve sleeve is uncovered, permitting escape of oil from the rod end of the horizontal table-propelling cylinder, so that the table will feed in horizontal direction. When a sharp slope is encountered, deflection of the tracer finger will take place. This deflection produces an upward component of the servo valve, which throttles off the discharge from the piston-rod end of the table cylinder. A vertical slope will cause sufficient deflection to close this port entirely and stop horizontal movement.

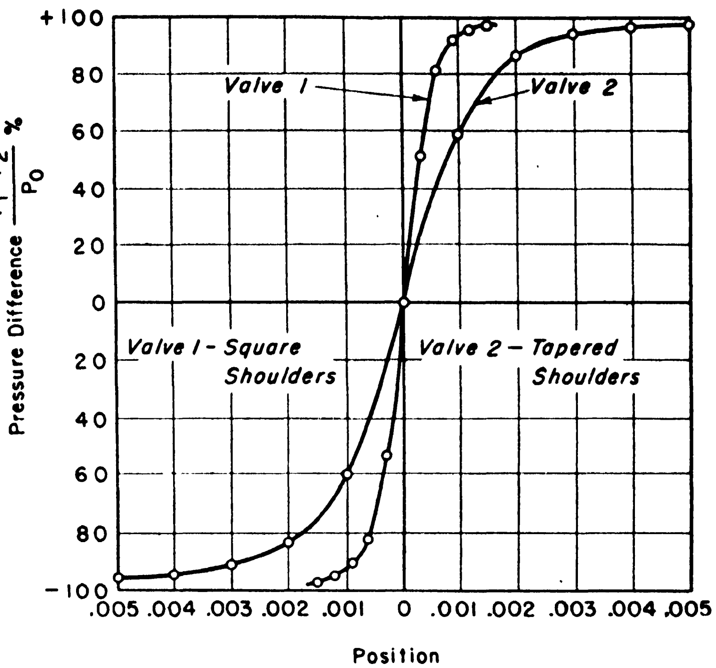


FIG. 252. Pressure difference vs. valve position in servo valve. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

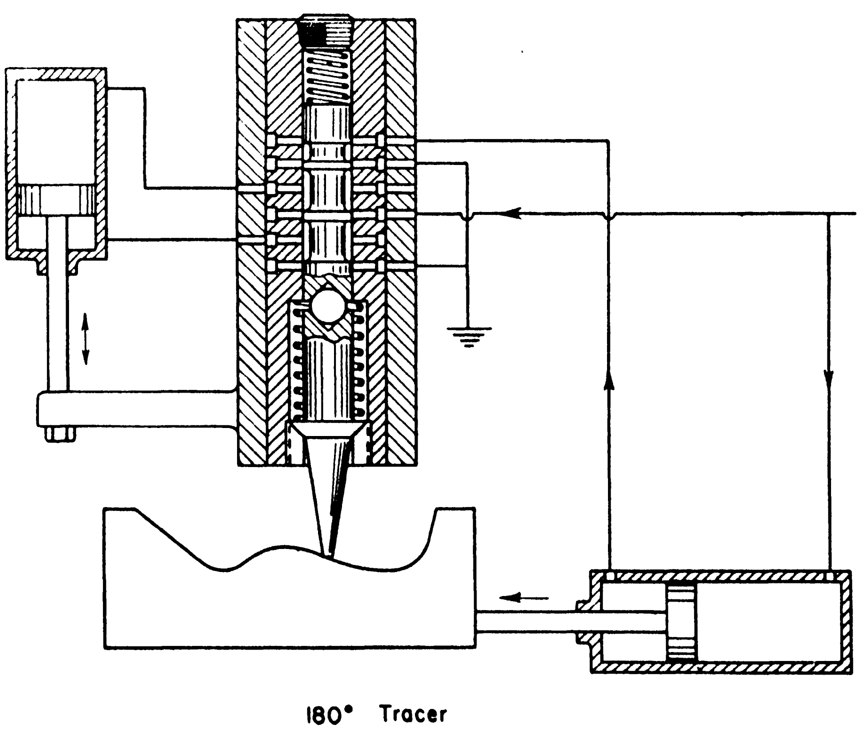


FIG. 253. 180° tracer. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

For 360° tracing, an ingenious device is used, shown schematically in Fig. 254. Here the axis of the tracer is placed perpendicular to the plane of the contour. The tracer servo mechanism is used as a control servo only, the feed components being produced by auxiliary valve-controlled power cylinders.

A handwheel (shown in Fig. 255) has the finger or tracer attached to it. Connected to this tracer is the servo valve, which controls the direction of rotation of a hydraulic motor. The latter is geared to the handwheel and to a cam, which controls directional-control valves for the two components of movement, namely, the table and ram movements. An arrow on the handwheel shows the direction of resultant movement between the cutter and the work. The tracer finger is mounted eccentrically with respect to the axis of the handwheel and may be displaced from its own axis. Three primary positions determine the direction of rotation of the tracer axis about the handwheel axis: underdeflected, undeflected, and overdeflected. In the underdeflected position, the tracer will rotate clockwise; in the overdeflected position, it will rotate counterclockwise, while in the undeflected position, no rotation will take place. The movement from one primary position to the next represents very small displacements (approximately 0.001 in.). In operation, the knob above the tracer is set in the "hand" position, and the arrow on the handwheel is pointed toward the template. As the tracer contacts the template, the mechanism shifts into the automatic position, and the tracing mechanism will immediately begin to seek equilibrium. If the tracer is overdeflected because of too much interference with the template, rotation away from template takes place to relieve this condition. This rotation changes the direction of the arrow on the handwheel and, therefore, the resultant direction of motion. If, on the other hand, the tracer is underdeflected, the rotation will be toward the template. While the tracer is seeking the correct direction, the slides are being moved hydraulically. In the extreme positions of the tracer, the rate of motion is reduced to provide smooth action around corners.

Figure 255 shows a machine set up for a milling operation that uses the 360° method of contour tracing.

4. Drilling, Boring, and Honing Machines. *The National Automatic Tool Co., Richmond, Indiana.* The multiple-spindle drilling machines built by this company constitute an excellent application of hydraulic power and control. Multiple-spindle adjustable machines are built in sizes up to 50-hp capacity. In addition, special-purpose production machines of the way type are built with and without indexing tables. In the following, their smallest multiple drills are shown and described. Figure 256 shows the G-5 high-speed sensitive multidriller and tapper.

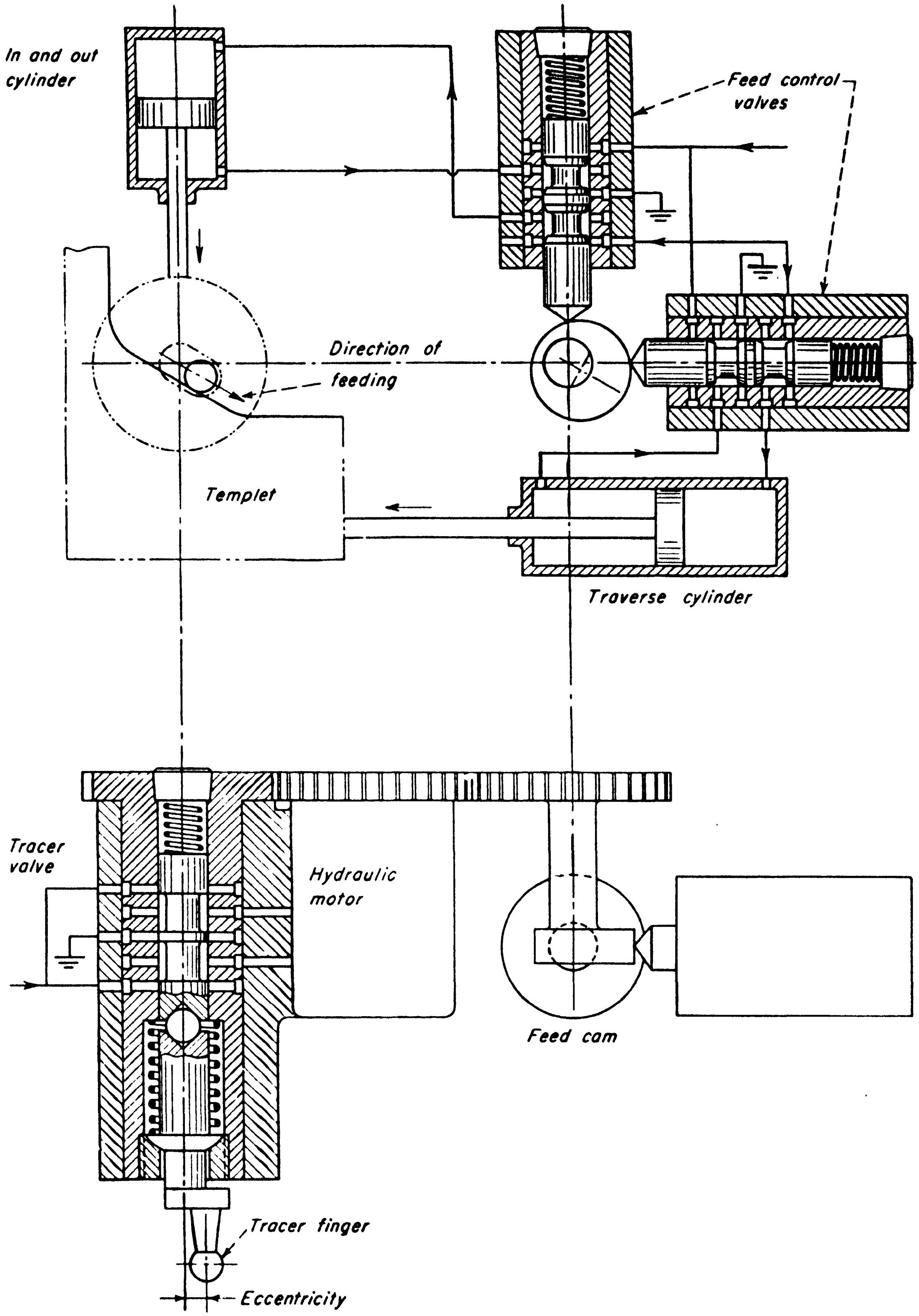


FIG. 254. 360° tracer. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

The machine has a stationary head and hydraulic-power-operated knee, which carries the table upon which the drilling fixture is mounted. The neck is of heavy cast-iron construction and is provided with two changes of speed and neutral. This change of speed with a ratio of approximately 2.3:1 is provided through a sliding gear arrangement actuated by an external lever located on the right-hand side of the neck. The neutral position is provided so that spindle rotation may be stopped for setup purposes.

FIG. 255. 360° tracer-controlled milling operation. (*Cincinnati Milling Machine Co., Cincinnati, Ohio.*)

Change gears are of the pick-off style and are mounted on splined shafts. These gears are located in the front upper portion of the neck under a cover.

The Natco Model G-5 may be furnished with a head for a 7- by 12-in. rectangular drilling area, and is provided with twelve ¾-in. diameter upper joint drives. The Natco Model G-6 may be furnished with a head for a 10- by 24-in. rectangular drilling area provided with 24 ¾-in.-diameter upper joint drives.

Natco adjustable spindles may be positioned as desired at any point within the drilling area of the head, provided the working angle of the adjustable drive does not exceed 35°. Operation of the table may be controlled through a manually operated lever located on the right front corner of the knee. Shifting the lever to the right places the machine in the semiautomatic cycle for routine operation. The table is then electrically controlled by a foot-operated treadle switch. Two adjustable

knobs and one transfer switch are located on the table knee. The upper knob is used to set the hydraulic valve for either a drilling or a tapping cycle. The lower knob is used to set the aperture or feed-rate valve. The transfer switch located on the right side of the table knee is used to change the electrical circuits for either a drilling or a tapping cycle.

FOR DRILLING ONLY. Normal drilling cycle is as follows after the operator depresses the treadle-operated switch: rapid traverse up, feed

FIG. 256. The Natco G-5 tapper and driller. (*National Automatic Tool Co., Richmond, Indiana.*)

up, rapid reverse to the starting position, and stop. In case of an emergency, the operator may reverse the table down to the starting position by depressing the treadle switch in the opposite direction. This cycle cannot be used for tapping operations.

FOR TAPPING ONLY. Normal tapping cycle is as follows after the operator depresses the treadle-operated switch: rapid traverse up, feed up, feed down, rapid reverse to the starting position, and stop. In case of an emergency, depressing of the treadle-operated switch in the opposite direction will cause the table to reverse its travel and return to the starting position and stop.

The table has a maximum feeding stroke of 6 in. Length of the rapid-

traverse and feeding stroke may be adjusted by setting the dogs that control the position of the spool-type valve. Length of feeding stroke is limited through an adjustable positive stop nut located on the lower end of the piston rod.

OPERATION THROUGH SERVO-VALVE LEVER. By shifting the servo-valve lever to the left, the operator may control the table cycle manually. The table may be traversed up or down by merely lifting or lowering the lever as desired. The rate of traverse is determined by the speed with which the operator raises or depresses the lever. The table will remain stationary at any point, provided the lever remains stationary. This feature is particularly desirable for setup purposes.

The G-5 machine has a drilling capacity of 8 holes, 5⁄16 in., 10 holes, 1⁄4 in., or 12 holes, 3⁄16 in. when drilling cast iron at 70 ft per min at 2 in. feed per min. Tapping capacity is 7 holes, 5⁄16 in. to 12 holes, No. 8–32 in cast iron. Head motor is 2 hp, 900 rpm, and hydraulic pump motor is 1½ hp, 1,200 rpm. The maximum feeding pressure is 1,500 lb, and feed rate is adjustable from ½ to 30 in. per min. Rapid-traverse rate is 200 in. per min.

The hydraulic circuit of the machine is shown in Fig. 257. Hydraulic pressure is supplied at the point indicated "supply." The hydraulic pumping unit for supplying the pressure consists of a box-type reservoir enclosing a pressure pump, relief valve, etc., and is bolted to the rear face of the column at the floor line. A tandem constant-volume vane-type rotary pump supplies the pressure to the hydraulic system. One portion of the pump supplies a high volume of low-pressure oil, which is used for rapid traverse. The second supplies a low volume of high-pressure oil for the feeding pressure. A metallic filter is provided to prevent dirt and grit from entering the hydraulic system. A pressure gauge of rugged double Bourdon tube, bushed-movement construction, together with a vibration protector, is furnished. This gauge enables the operator to check the pressure in the hydraulic system accurately at all times. A filler opening protected by fine-mesh screening is located in the top of the reservoir casting, and a visible oil-level gauge is provided on the right-hand side of the casting, by which the operator may determine the amount of oil in the reservoir.

The operation of the hydraulic system will be described in the following:

DRILLING CYCLE. Functions of the units during a drilling cycle are as follows:

1. Rapid advance. Press "start" push button. This starts hydraulic pump and head motor. The operator actuates the "forward" pedal of the foot-operated switch. This energizes the pilot-valve solenoid, shifting a spool-type pilot valve against spring pressure. This action

opens the ports from the pressure supply to the main directional valve. The oil pressure then moves the valve piston of the main directional valve against a positive stop plug. The ports of the valve are then opened from the pressure supply to the bottom end of the cylinder. The table then moves at rapid forward. The oil from the top of the cylinder is open to drain through the main directional valve and feed valve.

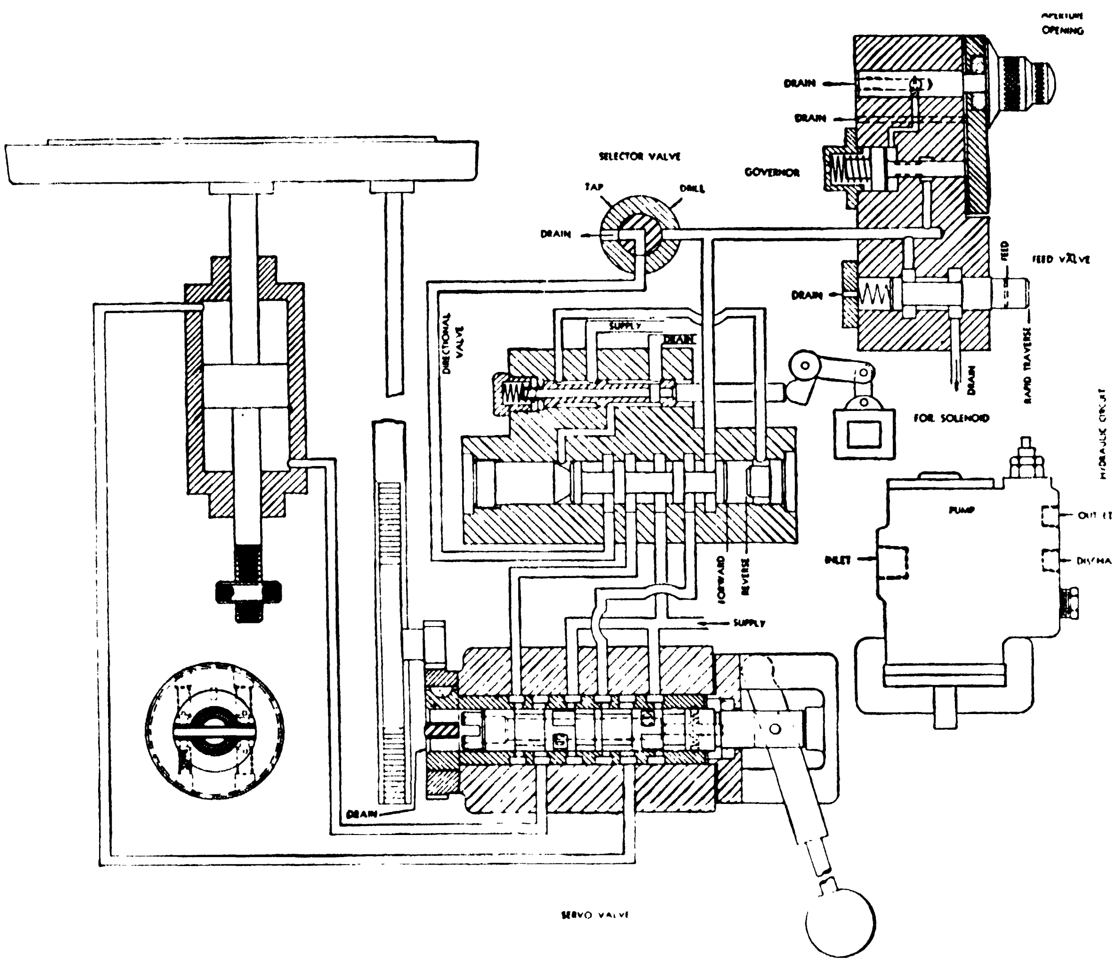


FIG. 257. Hydraulic circuit of G-5 Natco tapper and driller. (*The National Automatic Tool Co., Richmond, Indiana.*)

2. Feed. When the feed lever contacts the feed dog, carried by the table, the table is moved upward at a controlled rate of speed. The feed lever moves a valve piston of the feed valve against spring pressure to a position to block off the return oil from the top of the cylinder. This oil must then flow through a pressure-reducing-valve governor and into the metering aperture valve. The pressure-reducing valve maintains a uniform pressure differential to prevent the feed rate from varying as a result of the fluctuation of thrust loads developed during drilling cycle. The feed dog must contact the feed lever during the entire feed stroke. If it is released, the table will "rapid-advance" upward again.

3. Rapid reverse. When full depth of the hole is reached, a dog carried by the table contacts the reverse limit switch, which deenergizes the forward solenoid, releasing the pilot control valve. Spring pressure shifts the spool to the extreme right of the pilot-valve body, causing the ports from supply line to open to the reverse end of the main directional valve. The supply now shifts the main directional valve to position for rapid reverse. This blocks the supply to the bottom of the cylinder and directs the supply to the top of the cylinder. The main directional valve also opens the ports from the bottom of the cylinder to the selector valve and into drain. The table then moves off the feed dog, thus resetting the feed valve for the next cycle.

Tapping Cycle. In "tap" position the selector valve causes the oil in the cylinder to be metered through the same aperture and governor, while feeding to the work and away from the work. The cycle is started in the same manner as in the drilling cycle. Pressing "forward" pedal starts tap rotating and shifts the pilot valve to the "rapid-forward" position. Table then moves in rapid forward. Just before the tap enters the work, the feed dog contacts the feed lever, and then the table begins to feed forward and the taps enter the work. When taps have reached the desired depth, the dog carried by the table contacts the "reverse" switch. This switch deenergizes the pilot-valve solenoid and also reverses the motor. Taps then reverse, and table feeds down. Just after taps are disengaged from the work, the feed lever moves off the feed dog. The feed valve aided by spring pressure shifts to direct the supply line from bottom of the cylinder to drain. The table then "rapid-traverses" to down position and stops. The cycle may be repeated by pressing the forward foot switch.

Operation of the Servo Valve. The servo motor is a rotary unit that may be engaged or disengaged by shifting the operating lever up or down in the plane of the drawing and locking it in either position by the detent provided for this purpose. In the position shown, the servo control is inoperative, and the passages leading from both ends of the table cylinder connect to corresponding passages in the main directional valve. If the lever is raised in the plane of the drawing, the unit is in position for servo control. By turning the lever about the axis of the valve spindle (toward or away from the observer), ports in the valve register with corresponding ports in the sleeve, causing oil to flow from the supply port into one end of the table cylinder, and escape from the other end of the table cylinder into the exhaust, or vice versa, depending upon direction of valve movement. The sleeve may be rotated by rack and pinion from the table ram and constitutes the restoring or follow-up mechanism, so that the movement of the table will automatically follow

the movement of the valve lever as to speed, direction, and amount of displacement. Thus the table may be traversed at any feed up or down and stopped at any point by operating the servo-valve handle.

STEP DRILLING CYCLE. Functions of various parts during the cycle are as follows:

1. Rapid advance. Press "start" push button. This starts the head and hydraulic pump motors. The operator operates the forward pedal, starting the table to "rapid-advance." Simultaneously, the feed-dog positioning latch releases the feed dog by a solenoid action. A time relay is also started when the operator actuates the "forward" pedal of the foot switch.

2. Feed. At the end of the rapid advance, the feed-valve lever contacts the sliding feed dog, moving the valve to the feed-forward position. The feed portion of the cycle should be started a short distance before tools contact the work. At the start of feed, the feed-dog stop contacts the sliding feed dog. This stop accumulatively positions the dog for subsequent feed steps of the tools. After the tools reach a depth predetermined by the setting of the time relay, the forward solenoid is deenergized, allowing the valve to shift to the "reverse" position. The table reverses, moving down rapidly to the position where the initial feed started. At this point, the head "restart" switch is contacted, causing the "rapid-forward" solenoid to be energized. The table then moves at rapid forward again until the sliding feed dog contacts the feed-valve lever. As the feed dog was positioned at the point of reversal, the feed does not start until the tool point is a preset distance from the bottom of the previously drilled hole. This distance is the same as the point-to-work distance when the initial feed started. Simultaneous with the head restart, the time relay is again started. The cycle is repeated until full depth of the hole is reached.

3. Rapid reverse. When full depth of hole has been reached, as determined by the adjustment of the positive stop nut, a final reverse switch is contacted by a dog carried on the reverse-dog bracket. This switch deenergizes the forward solenoid, solenoid for the feed-dog positioning latch, and makes the "restart" switch ineffective. The table will then "rapid-reverse" to bottom of stroke and stop. The positioning latch returns the sliding feed to its initial starting position, and cycle has been completed.

4. Emergency reverse. The upward movement of the table may be reversed at any time during either the drilling or tapping cycles, by operating the "reverse" lever of the foot-operated switch. This deenergizes the pilot-valve solenoid, directing the oil to the top of the cylinder and opening the bottom of the cylinder to drain.

The Moline Tool Co., Moline, Ill. This company specializes in hydraulically actuated honing machines. Honing is a stock-removing and surface-finishing process combined in one operation, with the amount of stock to be removed being determined by the quality and accuracy of the surface that is presented to the honing tool. Figure 258 illustrates this company's No. 15 vertical, single-spindle, hydraulically reciprocated

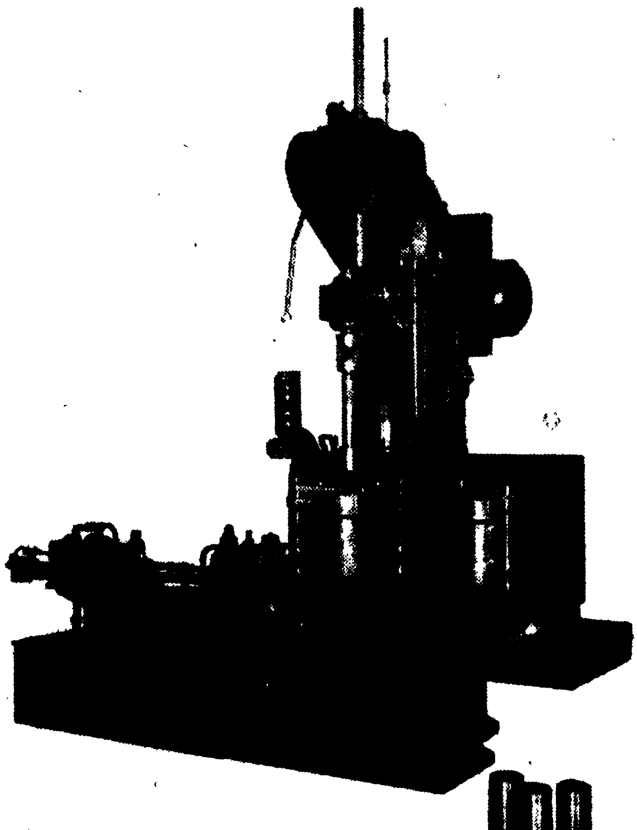


FIG. 258. Number 15 single-spindle hydraulic hone. (*The Moline Tool Co., Moline, Ill.*)

honing machine. The machine is driven by an electric motor, which provides power for the rotation of the honing tool as well as for driving the hydraulic pump. The motor runs continuously while the machine is in operation, but the hone rotation is started and stopped by a clutch, which is operated by a lever convenient to the operator. However, Moline hones are so constructed that they may continue to rotate, if desired, when withdrawn from the work.

HYDRAULIC RECIPROCATION. The slide on which the spindle is mounted is reciprocated on vertical ways by a hydraulic cylinder, and the rate of travel is set by the adjustment of a volume-control lever on the hydraulic control panel. In normal operation, the length of stroke is determined by the locations of adjustable trip dogs on the machine slide.

However, where it is desired to take short strokes for corrective honing, a convenient lever permits manual control to be used for producing these strokes at any point within the limits of the total travel as governed by the trip dogs. The reciprocation is started by electric push-button control and continues until stopped either by the automatic or the manual hone-control mechanism, whichever is furnished on the machine.

The hydraulic circuit of the machine is shown in Fig. 259. Power is

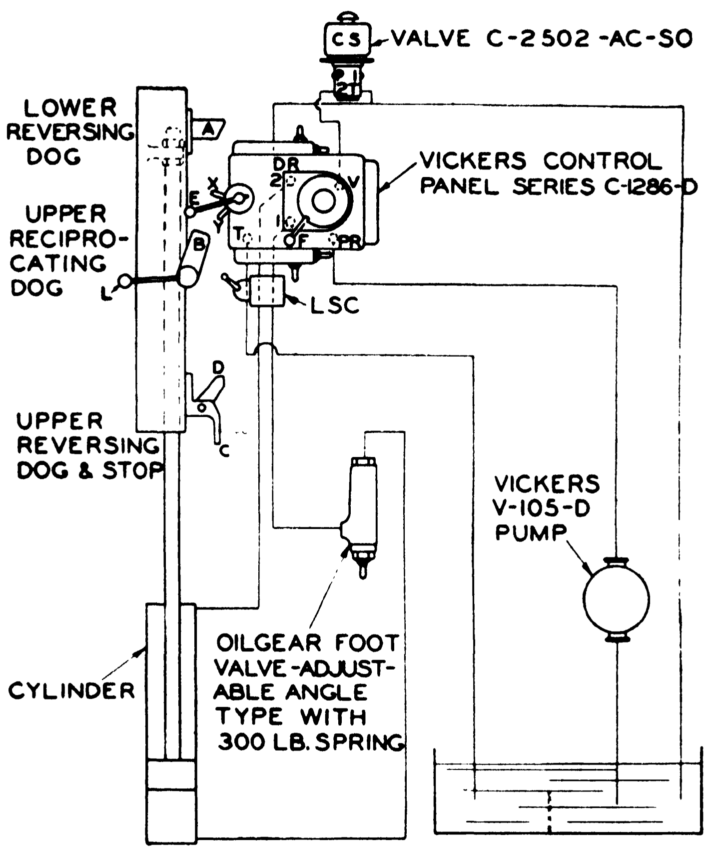


FIG. 259. Hydraulic circuit of No. 15 hydraulic hone. (*The Moline Tool Co., Moline, Ill.*)

supplied by a Vickers V-105-D pump, having a capacity of 10 gpm. A Vickers reciprocating control panel governs reciprocation of the hone by means of a pilot-valve-actuated directional-control valve. A meter-in type of flow control is built into the unit, permitting constant flow of oil to the reciprocating ram, regardless of resistance. A foot or resistance valve is placed into the pullback line to ensure positive support of the moving weights. The unit will reciprocate between the limits of dog *A* and *D*, which engage double lever *XY* at both extremes of the stroke. By engaging auxiliary trip dog *B* by means of lever *L*, reciprocation will take place between the limits of dog *A* and *B*. An electrically actuated vent valve is provided, whereby reciprocation of the hone may be stopped at the end of the return stroke by engagement of limit switch *LSC*, or by

manually operated switch. The machine illustrated in Fig. 258 is equipped with an indexing fixture that is hydraulically operated. A close-up of the hydraulic power unit and mechanism for indexing and clamping of fixture is shown in Fig. 260.

The Ex-Cell-O Corp., Detroit, Mich. One of the best known of the products of this company is their hydraulic power unit. This is a self-

FIG. 260. Close-up of hydraulic indexing and clamping mechanism. (*The Moline Tool Co., Moline. Ill.*)

contained unit, having a rotating spindle, motor driven through pick-off gears and V belt, and a hydraulic feed with variable-delivery pump unit for stepless adjustment of feeds. The entire unit is a complete and self-contained "package," and one or more of these units may be mounted upon a suitable base or frame in any desired position, so that their combination will constitute a complete drilling or boring machine for standard or special purpose. The units are especially suitable for automatic drilling, reaming, counterboring, and spot-facing operations. They are also used as prime movers for equipment to do other operations such as milling and, with guided tools, may be used for boring. A single tool can be held directly in the spindle shaft, or with a multiple-spindle head attached to the quill flange, multiple operations can be performed.

Length of rapid-traverse and feed strokes are controlled with easily adjustable control dogs. Feed rates are infinitely variable within a wide range and are dial adjustable. Spindle speeds can be varied with change gears in some units, and by changing belts and pulleys on others.

With remote control, two or more units on a machine can be started simultaneously, in sequence, or in any desired cycle. Operation of the units can be controlled from a limit switch, relay, or push-button station. Units installed in inaccessible positions can be started by remote control. Starting can be delayed until the work is properly clamped, until locating or safety devices are positioned, or until the operating cycle of another unit on the same machine is completed.

The compact design of Ex-Cell-O hydraulic power units makes it possible to place them at close center distances, either in line or radially around a fixture. Small over-all unit dimensions permit small machine dimensions, thus saving material cost as well as floor space. The hydraulic systems are entirely within the unit housings, even in the smallest unit. Hydraulic actuation of the quill assures smooth performance and ample thrust throughout the operating cycle. Neither the spindle nor the quill accelerates when the tool breaks through the work. Feeding thrust is applied directly behind the quill.

Ex-Cell-O hydraulic power units are easily installed and just as easily moved when product changes require rearranging the machine. This feature of flexibility makes it possible to use the units over and over again and to spread the initial cost throughout years of production. Holes in the base are provided to bolt the unit to the machine base or column, and a central key slot in the base permits accurate alignment with the work fixture. Units may be mounted horizontally, vertically with the spindle nose down, or at any angle between these positions. On vertically mounted units, counterweights should be provided whenever tools might be damaged by the quill creeping down while the machine is idle.

Each hydraulic power unit is arranged for driving single tools or flange-type multiple spindle heads. Morse taper adapters, adjustable in length, are available for driving single tools. The spindle nose has a straight hole with a keyway and setscrew. Each quill has boltholes and an accurate pilot diameter for attaching flange-type multiple spindle heads or other machining units.

Three sizes of units are available, specifications for which are given in Table I.

Figure 261 illustrates the No. 21 unit. In the No. 21 unit the hydraulic pump is coupled directly to the drive shaft, and the spindle is driven through gears. The different spindle speeds are obtained by change gears

TABLE I

Machine data	Style number		
	20	21	28-A
Motor horsepower	¾, 1, 1½, 2	1, 1½, 2, 3	2, 3, 5, 7½, 10
Horsepower required for operating unit	½–⅝*	⅝	¾
Maximum stroke, in.	4⅛	4⅛	15
Spindle speed, rpm, choice of one	920–10,000	370–2,085†	73–1,277
Maximum feed thrust, lb	500	2,200	10,500
Maximum rapid-traverse thrust, lb	160	300	800
Maximum weight lifted without counterweight, lb	60	100	225
Rate of feed, in. per min:			
First feed	4–180	1–25	¾–35
Second feed	.....	....	¾–20
Approximate over-all length, in	15¼	22¼	38⅛
Width, in	4½	10 13/16‡	13⅞
Height without motor, in	10½	10 7/16	14⅝
Net weight without motor, approx., lb	78	230	740
Morse taper number:			
In optional single tool adjusting sleeve	1	1 or 2	3 or 4
In optional single tool extension sleeve	2	3	5

* Depending on speed of spindle and drive motor.
† Speeds to 3,065 rpm extra.
‡ 12⅜ in. with remote control.

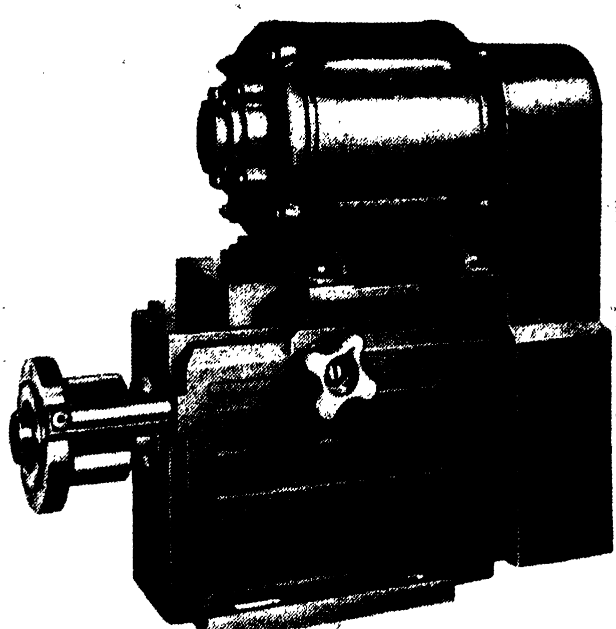


FIG. 261. Number 21 Ex-Cell-O hydraulic power unit. (*The Ex-Cell-O Corp., Detroit, Mich.*)

without affecting the pump speed. The variable-delivery-type pump delivers only the volume of oil necessary to overcome the machining resistance encountered by the quill. This results in efficient unit operation, because the temperature of the hydraulic oil is not raised excessively.

The length of the rapid-approach and feed movements and the reverse and stop positions are controlled by adjustable dogs. The operating cycle is started with the manual control knob, which may be mounted on either side of the unit, or with electrical remote control. The adjustable feed orifice is inside the remote-control knob. The automatic cycle includes rapid approach, feed, immediate or dwell reverse, rapid return, and stop.

The No. 21 unit is available with direct drive. In this unit the spindle and pump are driven independently through pulleys and V belts without the use of gears. This drive permits spindle rotation in either direction, allows changing spindle speeds conveniently, and results in exceptionally quiet operation.

Other extra equipment includes electrically actuated start and emergency return valve and controls for additional retraction of the quill. (The latter used when a short operating cycle is required for high production. Retraction of the quill beyond the normal stop position allows room for adjusting or changing tools.) The No. 21 unit may also be ordered for spindle speeds in excess of the standard unit.

The hydraulic diagram of the unit is shown in Fig. 262. The hydraulic system consists of pump *A*, sump *B*, cylinder assembly *C*, control-valve assembly *D*, relief and dwell-return valve *E*, dog assembly *F*, and orifice *G*. The remote valve assembly *H* is optional equipment. It is solenoid operated.

The hydraulic system is arranged to be self-scavenging. When a newly installed unit is started, a few strokes of the quill in rapid traverse will dispose of air accumulations. The hydraulic system is completely closed with no free breathing action between the sump and the outside. The unit operates most of the time with a slight vacuum in the sump. A check valve in the filler plug allows air to escape from the unit, but no air to enter. This prevents dust in the air from getting into the hydraulic fluid by either hydraulic action or by temperature changes.

The sump *B* is arranged inside the unit housing. Splash lubrication to every moving part in the unit is provided by this arrangement. The pump *A* is of the orifice-controlled variable-delivery type, also called "back-pressure-controlled variable-delivery type." It delivers only the amount of hydraulic fluid necessary to perform the required operation. The displacement is determined by the angular position of the swash plate *A*5.

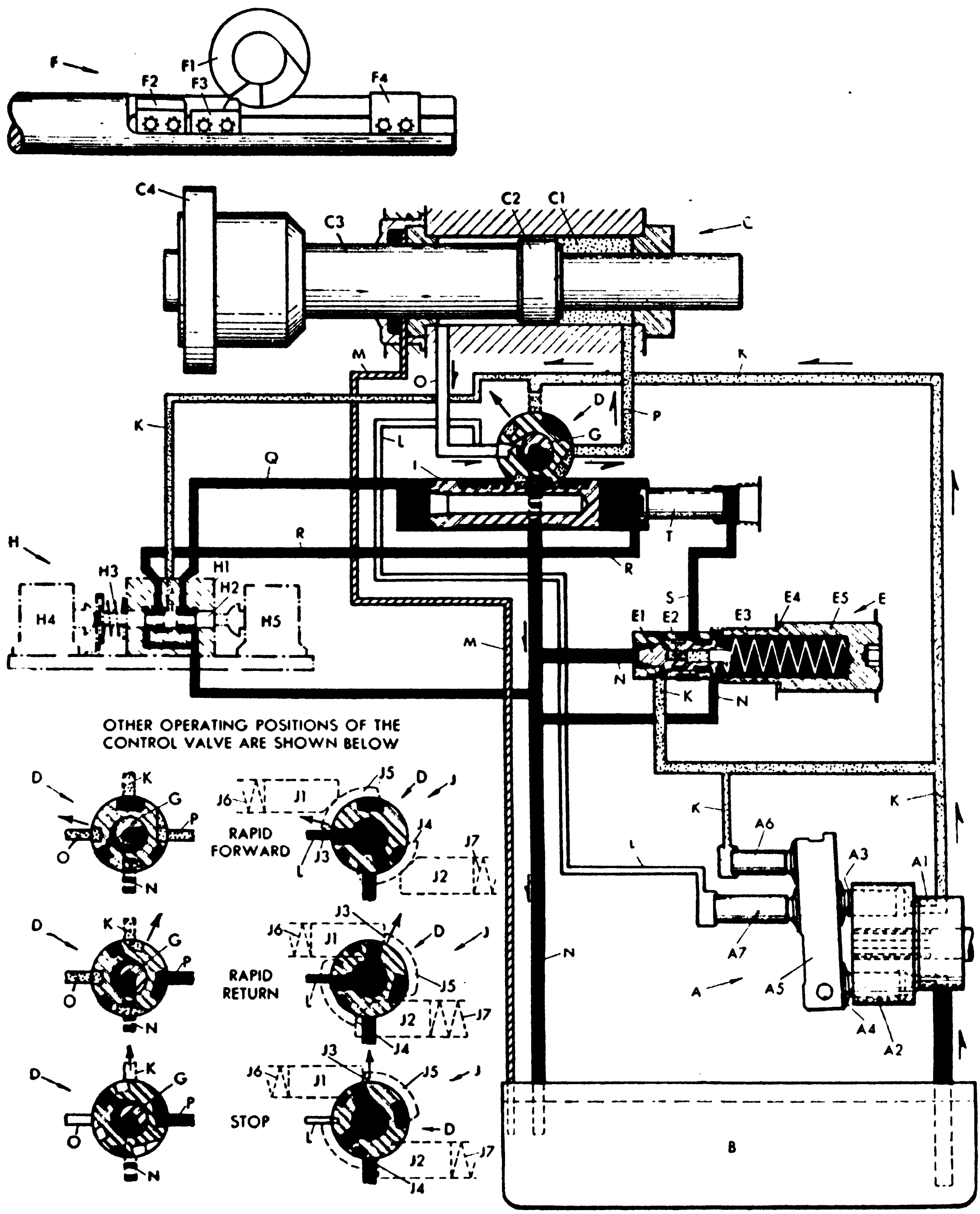


FIG. 262. Hydraulic diagram of No. 21 unit. (*The Ex-Cell-O Corp., Detroit, Mich.*)

The piston *C*2 is of the differential type. The rear area is twice as large as the front area. The remote valve assembly *H* is of the three-position type. The plunger is shifted by the solenoids *H*4 and *H*5 to the two side positions. It is centered by the action of spring *H*3.

There are two pressure systems in the circuit. These are pump pressure and pump-control pressure.

PUMP PRESSURE. The pump delivers fluid into the main pressure line *K*. During operation, the pressure in this line depends on the resistance encountered by the quill *C*3. The pressure may rise to the setting of the relief and dwell-return valve *E*. This valve may be set for pressures as high as 900 psi. The pump pressure may be checked by inserting a pressure gauge in the ¼-in.-pipe tapped hole at the left side of the gear box.

PUMP-CONTROL PRESSURE. The delivery of the pump is controlled from the pressure in the line *L*. During feed, pressure is maintained in line *L* by the return flow from the cylinder *C*1. During rapid traverse, line *L* is connected to exhaust permitting the swash plate to incline to the maximum position. While the unit is stopped, the line *L* is connected to the pump-pressure line *K*. The pressure in line *L* is not adjustable. It is approximately 140 psi during both feed and stop.

POSITIONS OF THE CONTROL VALVE. The control valve *D* has four positions. The positions in clockwise direction as seen from the dog side are

Rapid forward
Feed
Stop
Rapid return

The control-valve assembly *D* may be shifted manually within the limits of the setting of the control dogs to any of the positions. This is desirable while the unit is being set up. The selection of the positions is facilitated by lines marked on the control knob. During regular operation it needs to be shifted manually only to the rapid-forward position. It is shifted by the control dogs to the other positions. The control dogs engage trigger *F*1 on valve *D*. With remote control, the valve assembly *D* is shifted either to the rapid-forward or rapid-return positions.

In rapid forward the pump pressure in line *K* is connected to both the front and the rear of the piston *C*2. A high rate of rapid-forward traverse is possible by utilizing the return flow from the front of the cylinder on the rear area of the differential piston *C*2. The pump control line *L* is connected to exhaust.

During feed, the pressure in line *K* is connected to the rear of the piston *C*2. The front of the piston is connected to the pump-control line *L* and through line *O* to the feed orifice *G*. The rate of feed is determined by the setting of the feed orifice *G*.

The rapid return movement is started by reverse dog *F*4 shifting the control valve *D*. It is completed by the action of the valve return assembly *J*. As soon as spring plungers *J*1 and *J*2 engage the axial faces *J*3 and *J*4 in return member *J*5, the valve is shifted quickly to the

rapid-return position. During rapid return, the pressure is connected to the front of the piston $C2$. The rear of the piston is exhausted. The pump-control line L is also connected to exhaust.

The dwell-return movement is started by the quill engaging a forward stop. The pressure in line O to the front of the cylinder $C1$, which was connected during feed to both the feed orifices and the pump-control line L, will drop. The drop in pressure in the pump-control line L causes the pump delivery to be increased. The increase in the pump delivery will raise the pump pressure in line K until plunger $E1$ is shifted against spring $E3$. This permits pressure to pass into both line N and line S. The pressure in line K is determined by the tension of the spring $E3$. The flow caused by the pressure connection into line S will pass to the dwell-return plunger T. Plunger T will shift toward the front. As soon as spring plungers $J1$ and $J2$ engage the axial faces $J3$ and $J4$ in return member $J5$, the valve D is shifted quickly to the rapid-return position described previously. As soon as the quill $C3$ starts the return movement, the relief and dwell-return valve E closes, and the space behind plunger T is exhausted through lines S and the branch line N connecting the relief and dwell-return valve with the sump B.

At the "stop" position of valve D, the line O to the front of the cylinder $C1$ is closed; the line P to the rear of the cylinder is open to exhaust, and the pressure line K is connected to the pump-control line L. Thereby a low pumping pressure is maintained that will not cause excessive heat in the hydraulic oil.

During remote control, the "remote" valve H performs the identical actuation that otherwise is performed manually. For remote forward, the pressure is momentarily connected through line Q to plunger I. The valve D is shifted to the rapid-forward position. For emergency return the pressure is connected through line R to plunger I, which causes valve D to be shifted to the rapid-return position.

5. Small Presses and Broaching Machines. *The Oilgear Co., Milwaukee, Wis.* This company, one of the pioneer builders of hydraulic pumps, controls, and machine-tool feeds, is building a line of presses and broaching machines employing their fluid power equipment. Presses are available in a range of sizes and styles and equipped with pump and control systems to fit any desired application. Figure 263 shows a goose .eck type of press, adapted for straightening of shafts and bars and equipped with suitable fixtures for this purpose. The machine is operated by means of a two-way pump circuit with servo control, and is hand or foot lever operated.

The oil circuit of the machine is shown in Fig. 264, in which the reader will recognize many of the features discussed in detail in preceding chap-

ters. The unit shown comprises a high-pressure pump of the plunger type and a low-pressure rapid-traverse pump of the gear type. An auxiliary gear pump for supplying control pressure and lubricating and supercharging the high-pressure pump is also provided. A two-way suction-reversing valve of standard Oilgear design is flanged to the bottom of the pump casing. The pump is shifted by means of the servo control, so that it may supply pressure to either the forward or the retraction area of the

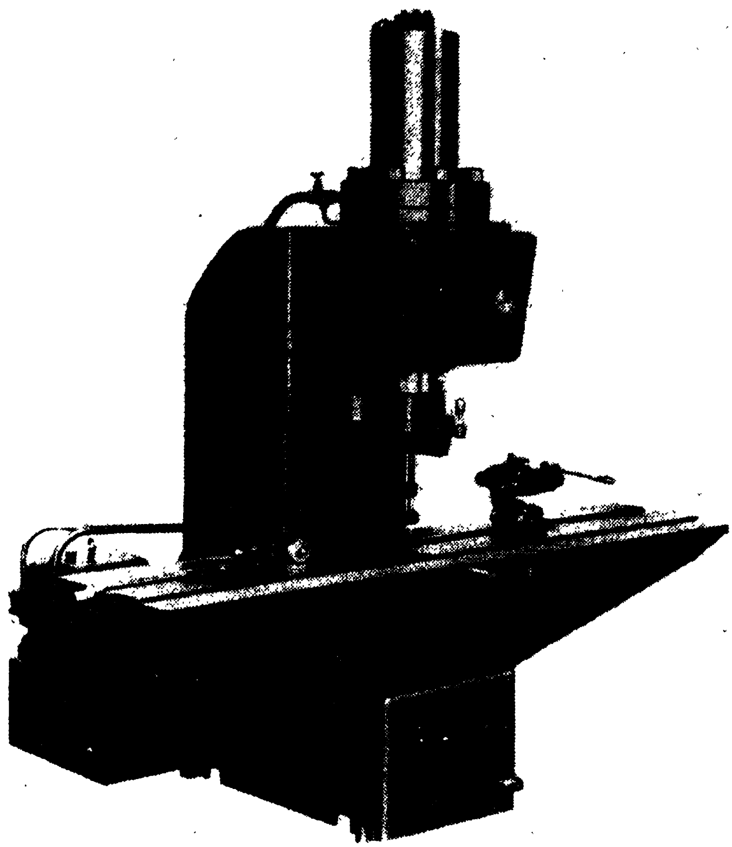


FIG. 263. Oilgear gooseneck-type straightening press. (*The Oilgear Co., Milwaukee, Wis.*)

ram of the press. The two-way suction, or differential, valve will operate in the manner described in the preceding chapter. The auxiliary or rapid-traverse pump is controlled in a unique manner. This pump discharges directly into the forward or down-pressure end of the press cylinder. When servo control is moved to "hard over forward," a small rotary pilot valve, which forms part of the servo valve, admits pressure from the auxiliary gear pump to the large piston of the rapid-traverse-pump unloading valve, shifting this valve to closed position, so that the rapid-traverse pump will discharge into the press cylinder. Should the pressure in the cylinder exceed 300 psi, the by-pass valve will be forced to the right against the auxiliary-pump pressure, and the rapid-traverse pump will by-pass. With the servo valve moved closer to neutral, the

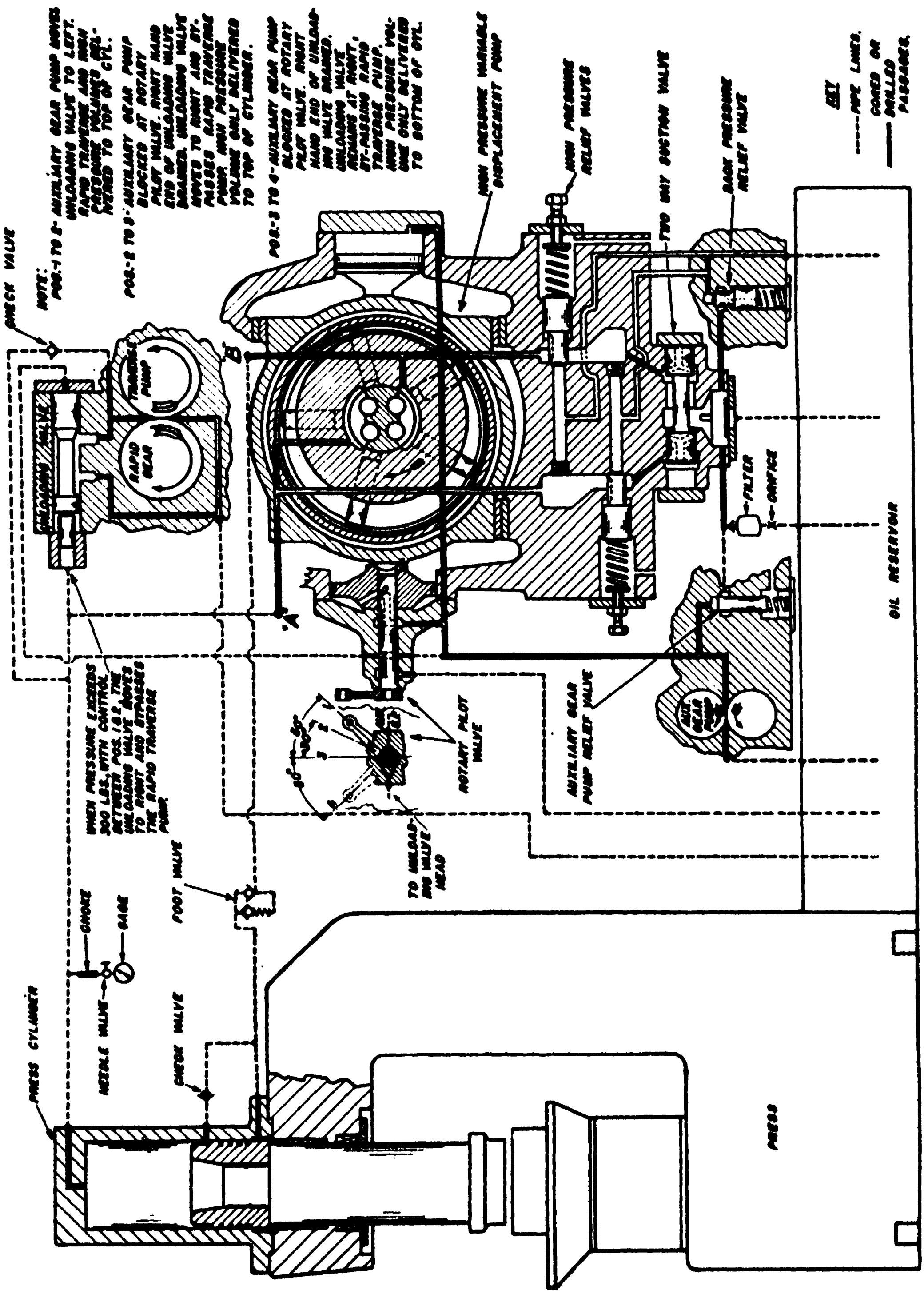


FIG. 264. Oil circuit of Oilgear press. (*The Oilgear Co., Milwaukee, Wis.*)

right end of the by-pass valve is vented, thus causing the valve plunger to shift and by-pass the rapid-traverse pump. The same condition obtains when servo valve is moved to the reverse position.

The Oilgear hydraulic broaching machine (Fig. 265), has developed from one of the pioneer achievements of this company. The machine employs a two-way reversing pump with regenerative differential circuit and the DX Oilgear pump, which has three controlled positions: forward, neutral, and reverse, which are selected through a built-in double-solenoid-actuated spring-centered pilot valve. Broaching and return speeds can be varied independently to coincide exactly with the operator's

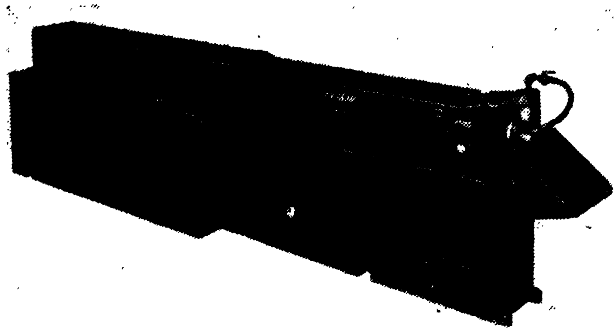


FIG. 265. Oilgear horizontal broaching machine. (*The Oilgear Co., Milwaukee, Wis.*)

ability to serve the machine. Accurate control of the broaching speed greatly increases the tool life, improves the finish and accuracy of the surfaces broached, saves power, simplifies the tool and fixture setup, and permits the use of maximum speed for the type of material broached.

The pump, motor, and reservoir are concealed in the machine base. Short pipe lines connect pump to differential valve and broach cylinder. Concealed wires connect push-button control to solenoids on pump. In operation, a push button is depressed to energize the lower solenoid on pump and start the broaching stroke of crosshead. A cam on crosshead contacts a dog on control rod to operate the micro switch, deenergize the solenoid, and stop crosshead instantly at the preset stroke. Depressing the other push button energizes the upper solenoid on pump and starts the return stroke of crosshead, at high differential advance speed. Again, a cam on crosshead operates the simple control to stop crosshead instantly at a preset point. Operator is free to stop or reverse ram movement at will.

A vertical surface broaching machine of the company's make is shown in Fig. 266.

6. Planing and Shaving Machines. *The Consolidated Machine Tool Corp., Rochester, N.Y.; Newton Machine Tool Works.* The combination mill and shaving machine built by this company is shown in Fig. 267. The principle of this machine is to take a roughing cut with rotary face-

milling cutters, as the table moves in with hydraulic feed. As the table reaches the end of the milling stroke, the shaving heads move down automatically, and a thin cut is shaved from the surface of the casting, as the table comes forward by rapid traverse to the indexing position.

The machine, as illustrated in Fig. 267, may be arranged with one or more unit milling heads and the same number of shaving heads, or less,

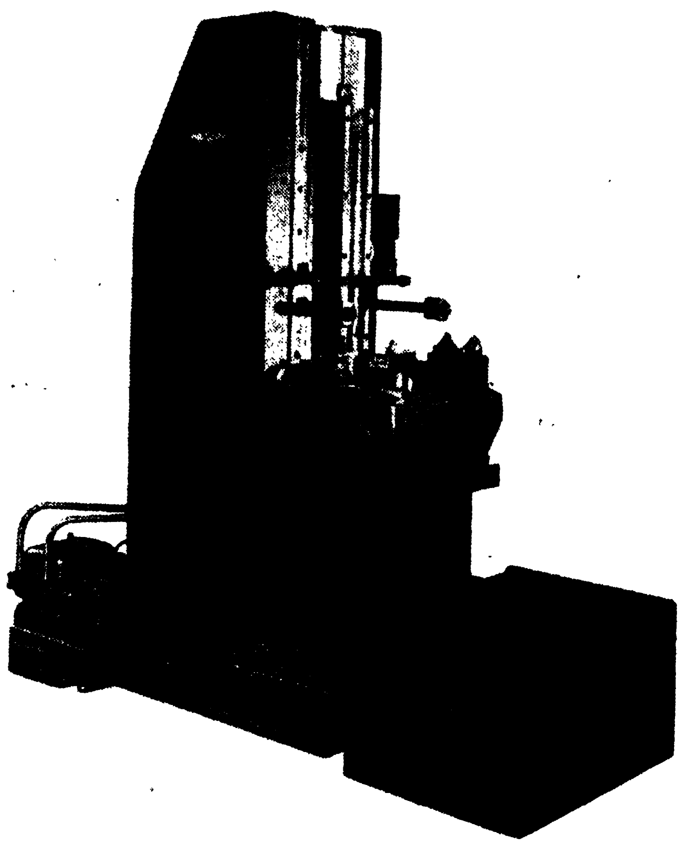


FIG. 266. Oilgear surface-broaching machine. (*The Oilgear Co., Milwaukee, Wis.*)

depending on the requirements of the work. The heads may be arranged horizontally, or vertically as in Fig. 267, or placed at any angle to suit surface to be finished. The shaving head is a heavy tool block in which are mounted one or more tungsten-carbide blades, and the head is carried on a ram with micrometer adjustment provided for setting depth of cut. The machine shown in Fig. 267 is arranged for loading and unloading the work from one position by means of an indexing fixture. The shaving head is arranged to draw up out of the way automatically while the work is rough-milled and to resume the cutting position just before return rapid reverse takes place. The entire cycle, including both movements of the shaving-head ram and rapid traverse back to loading position, is automatic.

The operation of a machine similar to the one shown in Fig. 267 is as follows: Normally the position of the table for loading the work is as shown. The work is loaded and clamped into that half of the fixture farthest from the cutters, and the fixture is then indexed or rotated 180° against a stop. The fixture is mounted on antifriction bearing and is raised, lowered, and positively locked at the four corners by compressed air. The cycle is manually started by the operator, and during this cycle

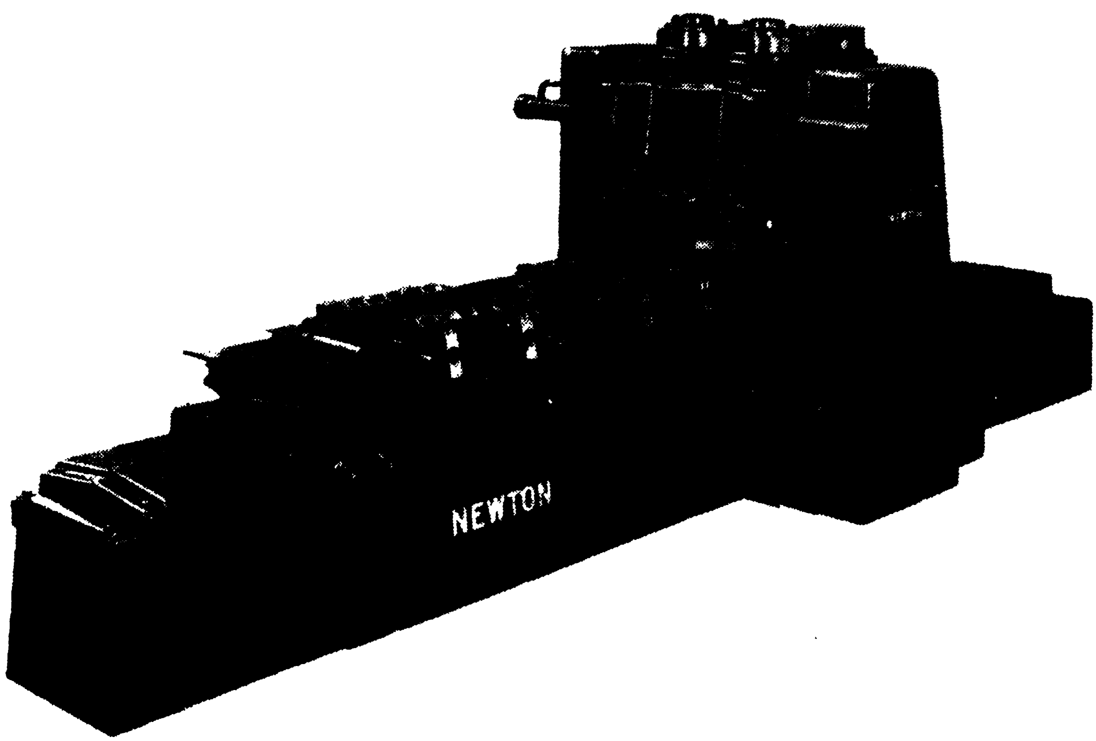


FIG. 267. Planer-type combined milling and shaving machine. (*The Consolidated Machine Tool Corp., Newton Machine Tool Works, Rochester, N.Y.*)

the other half of the fixture is loaded. As the cycle starts, the shaving head is automatically drawn up to allow the work to pass to the rough-milling cutter, and table advances in rapid traverse and drops into feed as the work reaches the milling cutter. At the end of the rough-milling cut, the shaving head advances to the cutting position, and table returns in rapid traverse to the loading position. It is during this rapid-traverse-return stroke that the work is shaved, producing a very smooth and accurate surface. While the shaving is in progress, no other machining is being done on the work. The machine is equipped with Vickers and Oilgear hydraulic units to obtain full automatic operation in the desired sequence. The hydraulic circuit is shown in Fig. 268 and will be described below.

Hydraulic power is supplied by a Vickers double pump, having a rapid-traverse pump of about 30-gpm capacity at 1,000 psi maximum pressure, and a feed pump of 5 gpm. The rapid-traverse pump is used for the shaving stroke and during this stroke develops more pressure than that required for the feed stroke. Speeds of about 14 ft per min are available for the shaving and 2 ft per min for the milling operation. The unit is operated by a 20-hp motor.

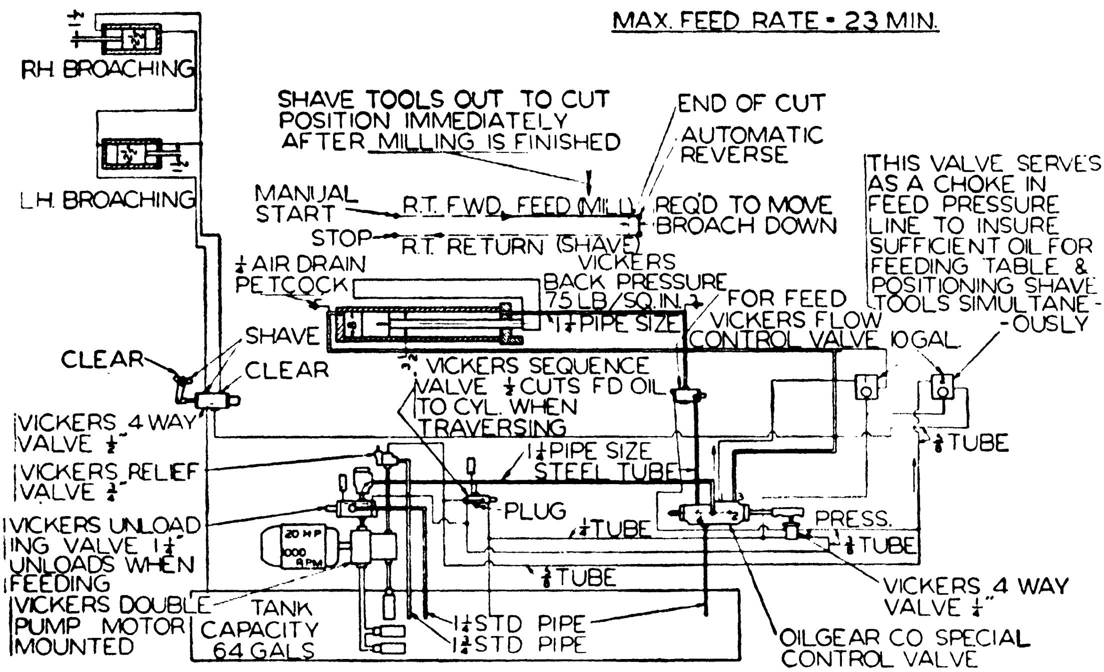


FIG. 268. Hydraulic circuit of Newton milling and shaving machine. (*The Consolidated Machine Tool Corp., Newton Machine Tool Works, Rochester, N.Y.*)

An Oilgear combination directional-control valve is provided which has five positions in the following order:

Rapid traverse return

Stop

Feed 1

Feed 2

Rapid traverse forward.

Feed 2 is not used in the circuit illustrated. Subsequent positions of the valve are initiated by dogs on the table, and reversal is obtained by a load-and-fire mechanism. In the "stop" position of the valve, the pressure port is open to tank, and the rapid-traverse pump will by-pass. A cam-operated plunger-type pilot valve is actuated by the movement of the main directional valve, and in its stop position permits discharge from the feed pump to open the sequence valve, whereby the feed pump will discharge into the common intake header and also by-pass into the tank.

Thus the machine is at rest with both pumps by-passing. The operator initiates the cycle by putting the main valve into rapid-traverse-forward position. When this is done, the cam on the stem of this valve releases the roller on the pilot valve, and the feed-pump pressure is directed into the pilot line, opening the back-pressure valve so that there will be no back pressure on rapid traverse forward.

Rapid-traverse pump discharges through line 3 into the head end of the table cylinder. At a preset position, the main control valve is shifted into feed position. Simultaneously, the cam on this valve depresses the plunger on the pilot valve and admits pilot-pump pressure to the sequence valve as well as to the unloading valve, opening both valves, thereby by-passing the rapid-traverse pump and permitting the feed pump to discharge into the main pressure line, through directional valve into line 1 and from there through meter-in valve 1 into the head end of the table cylinder. At the end of the feed stroke, the table actuates four-way valve to advance the shaving-tool pistons. Metering valve 2 apportions the oil to these cylinders to time the relative time of advancing tools and moving table. After completion of the forward stroke, the table trips a load-and-fire mechanism, which throws the main control valve past the stop position into rapid traverse. On rapid return, the rapid-traverse pump delivers oil into the retraction area of the table cylinder through line 4. Pilot valve is again released, and the feed-pump pressure is holding the back-pressure valve open. Upon completion of the return stroke, the table again actuates the four-way valve, and the feed pump retracts the shaving tools. Finally table comes to rest by moving the main directional valve to stop position.

The Rockford Machine Tool Co., Rockford, Ill. This company is one of the pioneer builders of hydraulically actuated machine tools. Their line of manufacture comprises hydraulically powered shapers in sizes ranging from 16 to 28 in., and also a 12-in. high-speed shaper and a 36-in. open side shaper of unusual design. Slotters are made in 12-, 20-, and 36-in. stroke, and both open-side and double-housing planers, all with hydraulic drive, are available. A development in which this company specializes, is the Hy-Draulic shaper-planer, a versatile machine tool that occupies a position somewhat between a planer and a shaper. The shaper-planer is a distinctive type of machine tool that combines the speed of a shaper with the accuracy and convenience of a planer. Its open-side construction adds to its productive capacity and ability to pay for itself quickly by handling most economically a wide variety of work, some of which would otherwise be costly to machine. The shaper-planer handles with ease many kinds of work that would tax the capacity of large shapers, owing to the fact that when making long strokes, the shaper ram

has a tendency to deflect under the pressure of heavy cuts. This tendency increases greatly as wear develops in the ram guides. Much work that is too large for a shaper, but too small to be handled efficiently on a planer, can be machined speedily and economically on the shaper-planer.

The basic advantages briefly outlined above were first realized in a shaper-planer having mechanical drive to the table. This construction, however, had certain inherent disadvantages and limitations. The cutting speed of the table varied constantly from end to end. The mechanical table drive required more power than is actually necessary for removing metal and did not utilize the full capacity of the cutting tool. Structural limitations restricted the ratio between cutting speed and the return stroke. The employment of gears, links, and slides in the table drive introduced possibilities for the development of chatter and gear marks in the surfaces machined. Despite these handicaps, which are present to some extent in all mechanical drives, the mechanical shaper-planer quickly earned for itself a prominent place and an excellent reputation in the equipment of tool- and diemakers, railroad shops, and plants having quantities of work that taxed the capacity of their larger shapers or else tied up the time of planers that could be used to better advantage on larger work.

In searching for possible improvements in the design and construction of the mechanical shaper-planer, Rockford engineers investigated the use of hydraulic pressure for driving the table. This led to the introduction of the Hy-Draulic shaper-planer, in which the feeds as well as the table were driven by oil under pressure. The Rockford Hy-Draulic shaper-planer was one of the first machine tools to employ this type of drive. As compared to mechanical drives, some of the advantages secured by utilizing hydraulic pressure for table drive and feeds are as follows:

Practically constant speed and pressure throughout the entire cutting stroke.

Hydraulic drive applies power in a straight line close up under table.

Much higher return speeds are secured. The rapid reversals at each end of the stroke are smooth and shockless.

Maximum capacity of tool can be used, and ample pressure is provided to cut metal at any speed without wasting power.

Cutting speeds and quick return are independent of each other.

Cutting speeds and feeds can be adjusted exactly to meet the requirements of any particular work, because the rate of these movements can be varied infinitely up to the maximum capacity of the machine.

More metal is removed per horsepower.

There are no bearings, gears, links, or joints in the table drive to develop

chatter or transmit marks to the work. Hydraulic drive contains few fast-moving parts subject to wear.

Work clears the tool on the return stroke before feed takes place.

Hydraulic drive has no gears to strip, cushions the machine against

TABLE II. SPECIFICATIONS OF HY-DRAULIC SHAPER-PLANER

Size machine, in.	42	66	90	120	144
Stroke lengths, max., in.	45	69	93	123	147
Cutting-speed range, ft per min.	0–80	0–80	0–80	0–80	0–80
Cutting pressure, max., lb.	9,500	9,500	9,500	9,500	9,500
Return-speed range, ft per min.	10–150	10–150	10–150	10–150	10–150
Feeds to railhead, in.					
Horizontal	0.001–0.200	0.001–0.200	0.001–0.200	0.001–0.200	0.001–0.200
Vertical	0.001–0.080	0.001–0.080	0.001–0.080	0.001–0.080	0.001–0.080
Feeds, vertical, in.	0.001–0.200	0.001–0.200	0.001–0.200	0.001–0.200	0.001–0.200
Floor space, stand. width, in.	75 × 218	75 × 240½	75 × 265½	75 × 311	75 × 359
Weight, with elec. equipment,* net, lb.	18,000	20,000	23,000	26,000	29,000

* Electrical equipment required: One 20-hp main drive motor, one 1½-hp traverse motor, one special shaper-planer control panel, one pendant push-button station; all suitable for the installation and its current requirements.

shocks, and protects tools and machine parts against damage, if the table is stalled from any cause.

Controls are simple and setup time decreased.

The machine is available in five sizes, specifications for which are given in Table II and is illustrated in Fig. 269.

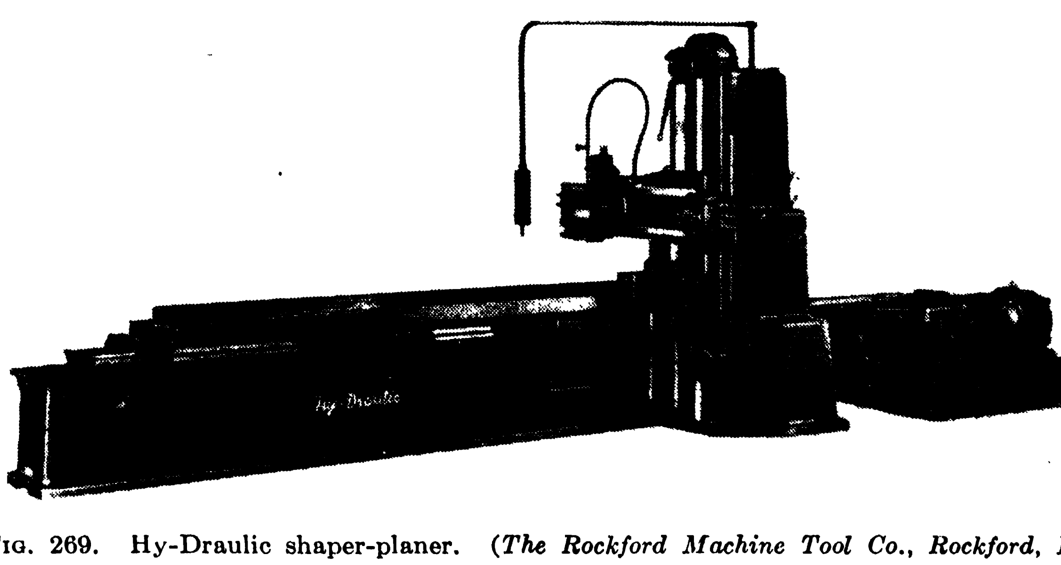


FIG. 269. Hy-Draulic shaper-planer. (*The Rockford Machine Tool Co., Rockford, Ill.*)

The piping diagram of this full hydraulic machine is shown in Fig. 270, supplied by courtesy of the Oilgear Co., Milwaukee, Wis. In this unit the table is attached to the piston rod of a hydraulic cylinder (illustrated in Fig. 135), and the flow of hydraulic oil to the cylinder is valved by

means of a hydraulically actuated three-way valve so that oil pressure is permanently supplied to the rod end of the cylinder, which is the power end for the cutting stroke, so that the table is pulled on the cutting stroke, putting the piston rod in tension. During this stroke the oil is exhausted from the large end of the piston through a spring-loaded back-pressure valve that prevents sudden lunging of the table in case of sudden drop of cutting resistance. On return stroke, oil pressure is supplied to the large end so that oil is supplied to both ends of the piston and the table is returned at high speed, corresponding to the volume of the pump acting

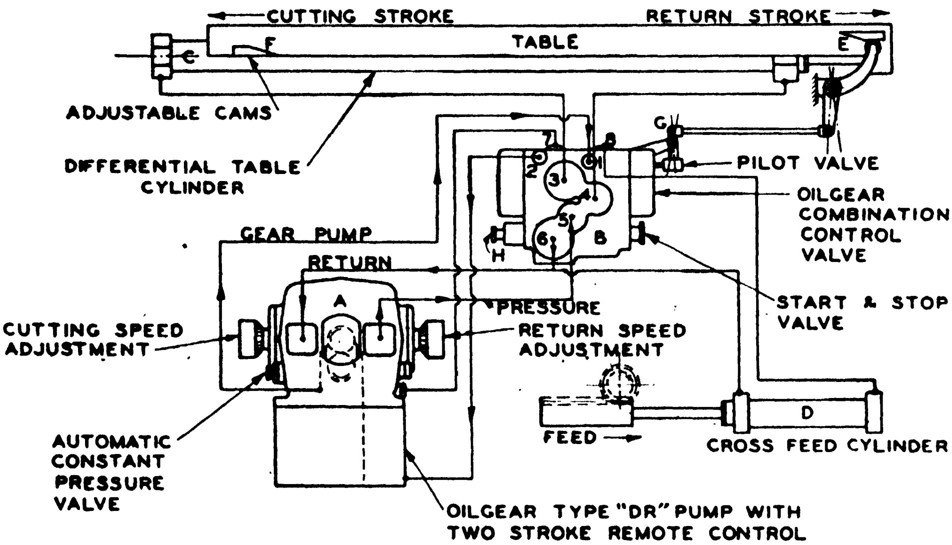


FIG. 270. Diagram of Rockford Hy-Draulic shaper-planer. (*The Oilgear Co., Milwaukee, Wis.*)

on the piston-rod area. The pump is an Oilgear Type DR with a control similar to that illustrated in Fig. 100. Two independently adjustable speeds are available for cutting and return. A pilot and supercharging pump keeps the entire system under a head of pressure, as customary in Oilgear systems, and supplies pilot pressure for actuation of the main directional-control valve by means of a four-way pilot valve, operated by dogs on the table. This permits continuous reciprocation of the table at adjustable speeds between adjustable reversing points. A start-and-stop valve, which is merely a by-pass valve between pressure and return line, manually operated, is provided. Great care is used in the design of the directional-control valve, which is pilot operated through suitable chokes to control carefully the rate of reversal in both directions. V-slotted ports in this valve further assist in obtaining extremely smooth and gentle reversing action of the heavy table at high speeds.

The cross-feed is operated from a double-acting cylinder by means of an overrunning clutch. The cylinder is tied into the hydraulic system, and the feed piston will reciprocate in unison with the reversals of the

main piston. By means of the overrunning clutch, feed stroke is obtained at one end of the table travel only.

7. Die-casting and Plastic Machines. *The Reed Prentice Corp., Worcester, Mass.* Hydraulically operated machinery made by this company comprises their line of die-casting and plastic-injection machines. Die-casting machines are made both in gooseneck and cold-chamber design for zinc, aluminum alloys, and brass. Requirements

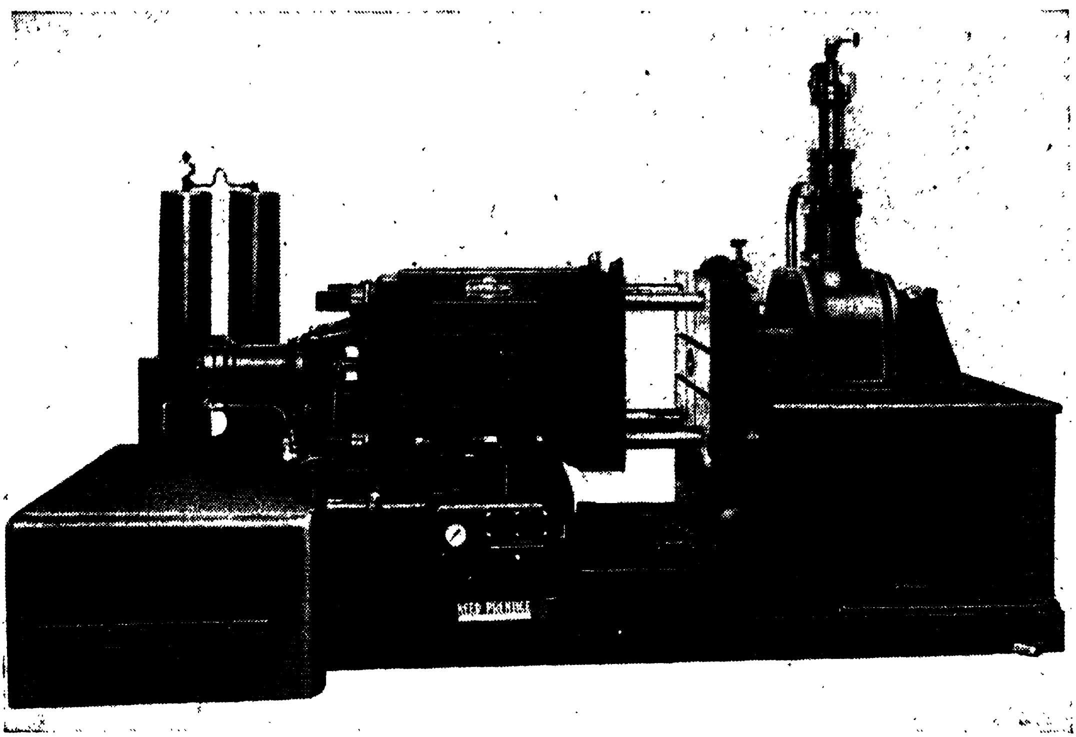


FIG. 271. Number 1½ die-casting machine. (*The Reed Prentice Corp., Worcester, Mass.*)

peculiar to die-casting practice are reflected in the interesting design of the hydraulic system of these machines. One of the most popular models is the No. 1½ machine, which, by an ingenious arrangement of parts, may be made into either a gooseneck machine for zinc, tin, and lead-base alloys, or a cold-chamber machine for aluminum, magnesium, and brass alloys. The machine, with holding furnace and gooseneck, is shown in Fig. 271.

In principle, the machine consists of a hydraulic cylinder actuated toggle mechanism that serves to clamp the two halves of the die-casting die securely together. After locking the die halves together in this manner, molten metal is injected by means of a hydraulically operated plunger. The molten metal is carried in a holding furnace shown at the right side of Fig. 271, and is permitted to enter a gooseneck-shaped tube mounted inside the furnace. A plunger directly above the gooseneck is operated by

hydraulic pressure and injects the fluid metal into the closed space between the dies. Complex dies are equipped with core-pulling cylinders, which pull cores before the finished casting is ejected after opening the machine by means of the toggle-operating cylinders.

The No. 1½ gooseneck machine has die plates 28 by 29 in., a maximum die opening of 18 in., and 10-in. stroke. The toggles have a locking pressure of 150 tons. With a 3¼-in.-diameter hydraulic piston and at 1,000 psi hydraulic pressure, 8,300 lb pressure are available. Metal plungers range from 1¾ in. to 3¼ in. in diameter, corresponding to 3,460 to 1,000 psi injection pressure. Zinc castings up to 10¼ lb in weight may be produced. The minimum time per cycle is 7 sec. The machine weighs about 15,000 lb.

For aluminum and brass alloys, the gooseneck scheme is not feasible, and molten metal must be ladled into the shot cylinder by hand ladle. To this end the machine is converted by removing the holding furnace and gooseneck and substituting a horizontal shot cylinder, hydraulically actuated. In all other respects, the machine is the same. The operation of the machine with cold chamber is as follows: The machine is arranged for both manual and semiautomatic operation. For semiautomatic operation, dies are closed manually by push buttons on the control panel, operator ladles the metal, then steps on the foot switch to shoot the plunger forward. The remainder of the cycle is automatic and is controlled by an adjustable electrical timing device. Plunger remains forward and when the dies open, will travel an additional 3½ in. to eject slug before returning. When manually operated, the four push buttons on the control panel are used. The upper two open and close the die; the next lower two actuate the plunger. The machine is electrically interlocked, eliminating all danger of shooting the metal until the dies are securely closed. Arrangements are made for automatic ejection of castings. Die plates are finished on side to provide for core-pulling attachment, and push button is provided to actuate core-pulling cylinders when they are used.

The aluminum machine has an injection-cylinder diameter of 6 in. and a total plunger stroke of 11 in. This with a 2-in. plunger produces 9000, psi metal pressure. Aluminum castings up to 3¾ lb in weight may be produced.

The hydraulic diagram of the machine is shown in Fig. 272. Power is supplied by a Vickers high- and low-pressure combination pump, supplying 34.6 gpm at 250 psi and 7.1 gpm at 1,100 psi. The unit is equipped with a combination unloading and relief valve, which has been described previously (see Fig. 160). Oil is discharged from this pumping unit directly to a pilot-operated four-way valve of conventional design for

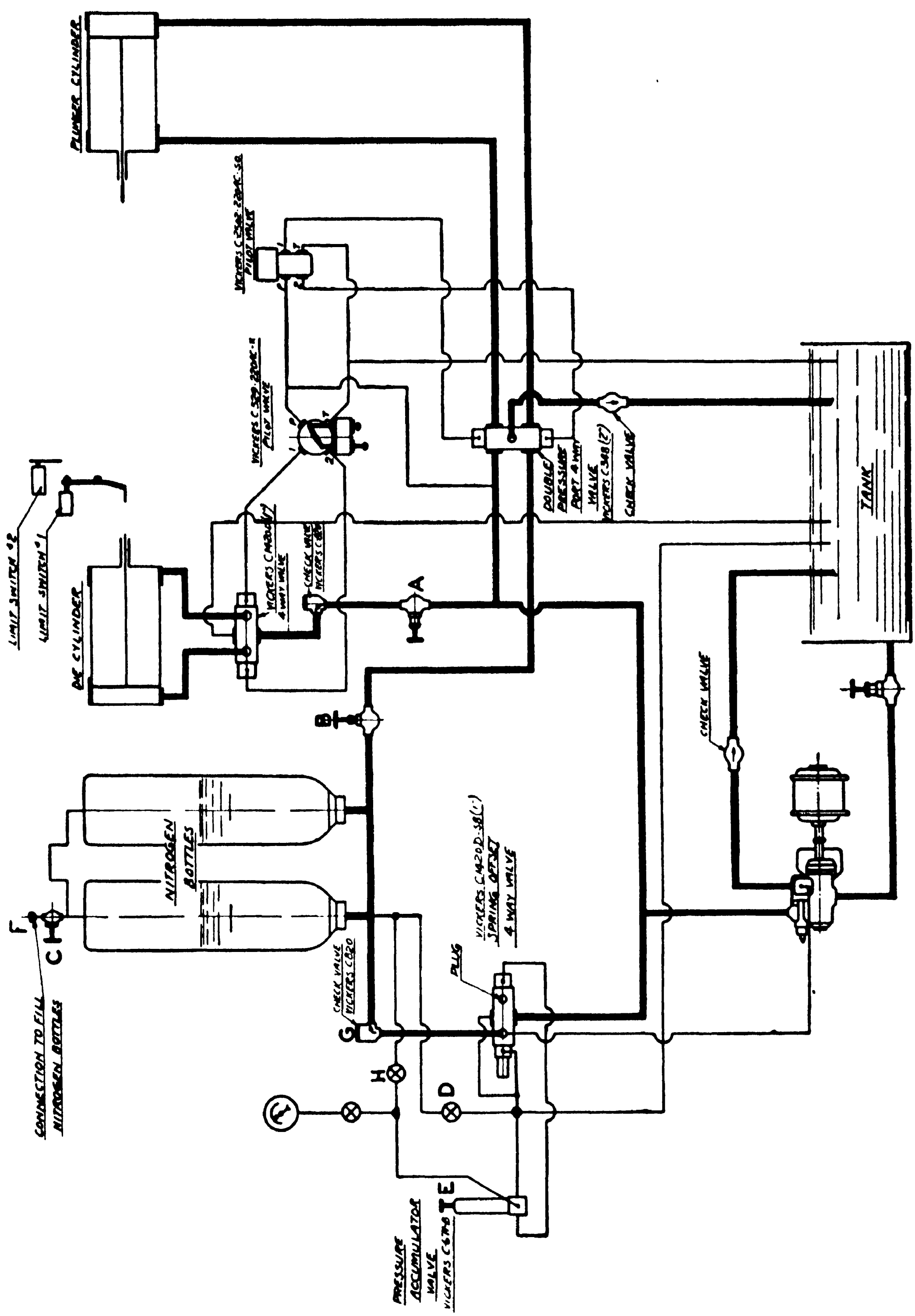


FIG. 272. Hydraulic diagram of No. 1½ die-casting machine. (*The Reed Prentice Corp., Worcester, Mass.*)

controlling the action of the clamp-operating cylinder. Another branch leads to one of the intakes of a double-pressure port four-way, which is so arranged that direct flow from the pumps is used for the retraction of the shot-cylinder piston. Oil for the forward or injection stroke is supplied by a second pipe line leading to nitrogen-loaded accumulators, which permit fast injection of metal by oil stored under pressure. The accumulators may be charged by the pumping unit at a pressure controlled by the spring-loaded pressure-operated sequence valve E. This valve, responding to the pressure existing in the accumulators, will actuate a

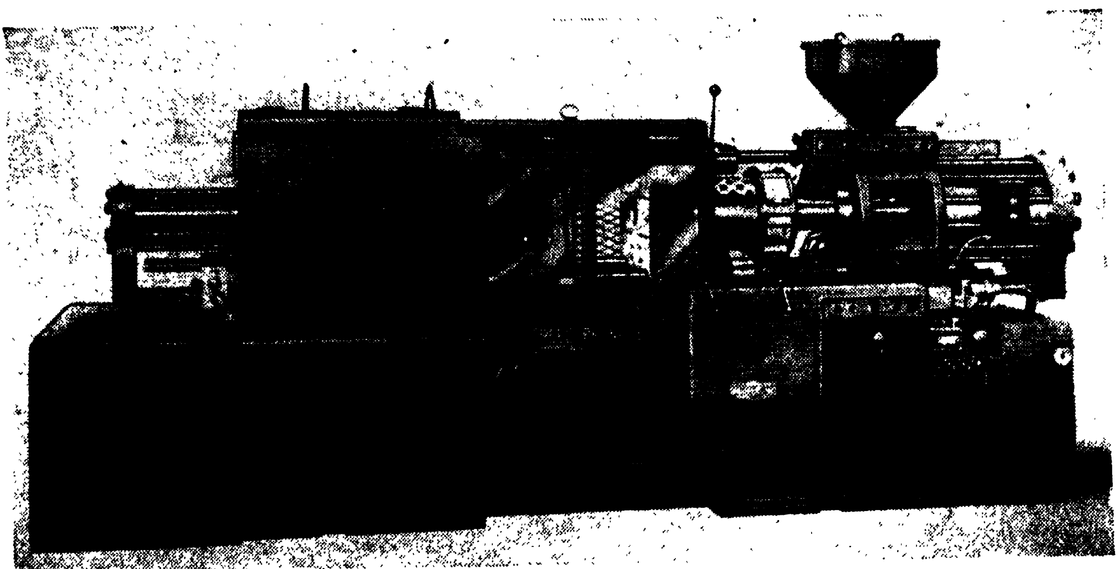


FIG. 273. No. 10 D8 plastic-injection machine. (*The Reed Prentice Corp., Worcester, Mass.*)

pilot-operated spring-opposed valve. In its normal or spring-actuated position, this valve closes off the connection between the pump-pressure line and bottles. Lowering of the bottle pressure below a preset amount causes the pilot valve E to admit pressure to the pilot end of the four-way valve, overcoming the spring tension and, in turn, connecting the pump discharge to the bottle. When this is done, a pilot line connected to the top of the unloading piston receives pressure and prevents the unloader from opening. Thus the full capacity of both pumps is available at full charging pressure during the short interval required to restore the charge of oil in the bottles. As soon as the pressure in the bottles has reached the preset amount, valve E will release pressure from the pilot end of the four-way valve, so that this valve will again close off connection between pump and bottles and release pressure on top of the unloading valve, permitting the large-volume pump to by-pass. Constant pressure on the entire system is maintained by the small-volume high-pressure pump. An oil cooler is supplied with the machine to care for the heat developed

in operation and while the small constant-delivery pump maintains pressure.

The plastic injection machine made by the Reed Prentice Corp. is shown in Fig. 273. The principle of a plastic-injection machine very closely resembles that of a die-casting machine, with the main distinction that instead of feeding molten metal into a cavity from which it is injected into the die, granular plastic material is fed from a storage hopper into an electrically heated plasticizing chamber from which the plasticized material is injected into the closed dies by means of a hydraulically actuated plunger. Generally, a high speed of injection is not required for

TABLE III

Ounces per shot	8
Plasticizing capacity per hour	45 lb
Diameter of injection plunger	$2\frac{5}{8}$ in.
Pressure on material	20,000 psi
Stroke of injection plunger	10 in.
Rate of injection	10.8 cu in. per sec
Size of die plates	21 by 25 in.
Mold opens	$10\frac{1}{4}$ in.
Maximum die space	16 in.
Mold closing pressure	250 tons
Combined capacity of high- and low-pressure pumps	60 gpm
Motor required	20 hp
Weight	15,000 lb

the plasticized material, so that pressure-storage bottles are not required. Plastic machines are made in capacities from a few ounces to several pounds. Specifications of the 8-oz Reed Prentice machine illustrated in Fig. 273 are given in Table III. The machine is equipped with a Vickers hydraulic system, illustrated schematically in Fig. 274.

Power is supplied by a combination high- and low-pressure unit to two pilot-operated four-way valves, which control the toggle clamp and injection cylinder. A reducing valve is provided that permits application of a controlled pressure to the injection cylinder.

8. Hydraulic Press Welders. A fertile field for application of hydraulic power is the so-called "press welder." The principle of this machine consists in a hydraulically actuated electrode that firmly presses the parts to be welded together against a stationary electrode and dwells while current is passed through these parts, which fuses them together almost instantly. Hydraulic power and control systems for these machines are simple, consisting essentially of a large-capacity pump to advance the traveling electrode rapidly, decelerating or cushioning valves to prevent impact at the instant that the parts contact, and a directional-control valve, cam-actuated, to permit operation at a predetermined

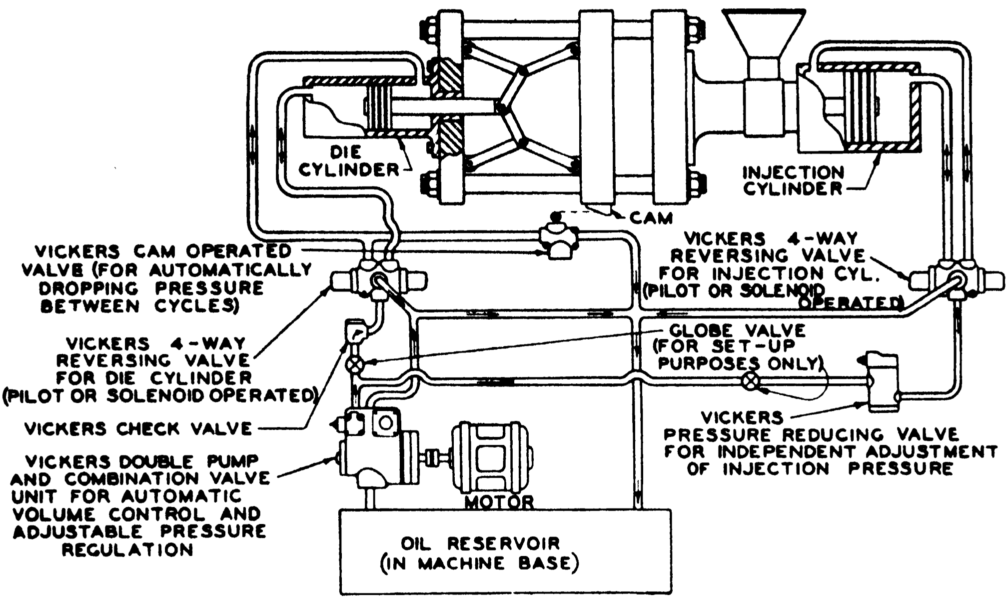


FIG. 274. Hydraulic operating circuit of 10 D8 Reed Prentice plastic-injection machine. (*Vickers Inc., Detroit, Mich.*)

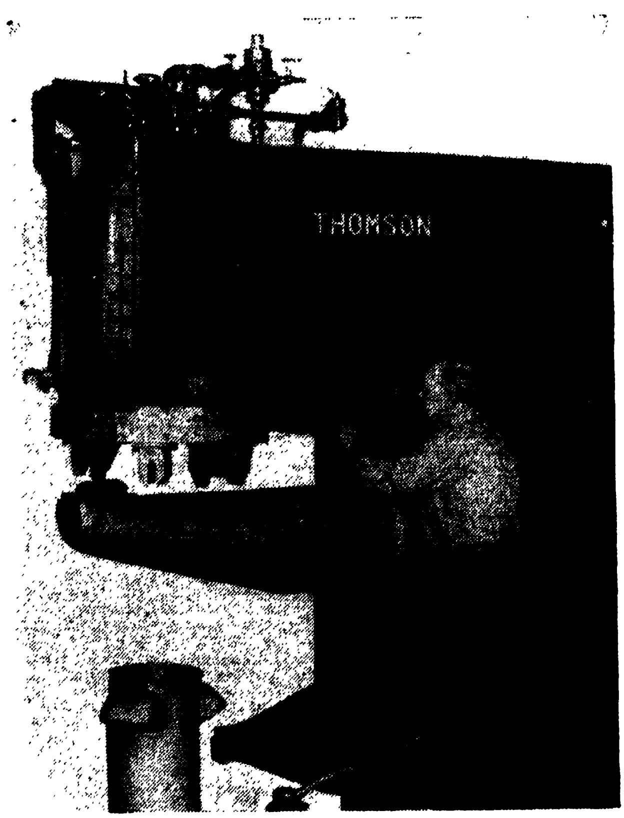


FIG. 275. Hydraulic press welder. (*Thomson Electric Welding Co., Lynn, Mass.*)

cycle. Hydraulic power is ideally adapted for this purpose, as it fills all requirements for rapid approach, smooth deceleration, dwell pressure and follow-up, and rapid separation and return.

A hydraulic press welder, made by the Thomson Electric Welding Co. in Lynn, Mass., is shown in Fig. 275.

The machines illustrated and described in the preceding paragraphs are a few representative samples of the possibilities of hydraulic power and control. New applications are coming into being every day. Machines of great complexity and with multiple automatic circuits are being developed, but the underlying principles are basically the same. If this text has contributed to the understanding of these basic principles and assisted the reader in their utilization, it will have served its purpose.

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